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TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 44

1922

PUBLISHED BY
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
29 West Thirty-ninth Street, New York

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Dexter S. Kimball

PRESIDENT 1922
OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 44

ATLANTA MEETING
NEW YORK MEETING
1922

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1923

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

To the Members:

In pursuance of a policy designed when the Society was organized, the wisdom of which years of experience seem to confirm, matter for presentation before the Society is first submitted to a Standing Committee on Meetings and Program. Subsequently, and in the light of discussion of such matter, an independently organized Committee on Publication and Papers determines what material shall have the further distinction of appearing in Transactions.

Contents of Volume 44 of Transactions

The papers and committee reports presented during 1922 before the Spring, Annual, Regional and Local Section Meetings have been painstakingly selected by the Committee on Publication and Papers with a view to their value for permanent record.

Morocco-Bound Copies

The annual volume of Transactions is issued free to members in paper binding. Half-morocco binding is furnished at cost for $1.50. Members desiring volumes bound in this way from year to year are encouraged to file such an order with the Secretary. This will be a convenience both to the member and to the administration of the Society. The additional cost will be charged to the member's account.

Important to Members Interested in Stress Analysis

At the 1922 Annual Meeting, under the auspices of the Railroad Division, an extended paper dealing with stresses in locomotive frames was presented by R. Eksergian. Since the Committee on Publication and Papers was compelled to limit the size of Transactions, this paper was not included in Volume 44, but on account of its extreme value, it was decided to print it in pamphlet form along with the discussion, and supply copies gratis to members whose requests are received by the Secretary before October 1. A synopsis of the paper will appear in the June and July issues of Mechanical Engineering, which members are invited to study.

During the war, owing both to government requests to limit the use of paper and to the high costs of production, the Society was prevented from maintaining its accustomed regularity in the publication of its Transactions. With this issue, making a second volume sent out this fiscal year, the Society is again up to its schedule. With an increase of almost 150 pages over volumes of preceding years, it has been possible to issue Volume 44 in more compact form, owing to the thinner paper stock which has been used.

Calvin W. Rice, Secretary
29 West 39th Street
New York

July, 1923

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

OFFICERS AND COUNCIL

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Dexter S. Kimball ..... Ithaca, N. Y.

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Terms expire December 1922

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Robert B. Wolf ..... New York, N. Y.

Terms expire December 1923

E. A. Deeds ..... Dayton, O.
Robert Sibley ..... San Francisco, Cal.
L. E. Strothman [1] ..... Milwaukee, Wis.

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Terms expire December 1922

Elbert Curtiss Fisher ..... Saginaw, Mich.
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Terms expire December 1923

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Terms expire December 1924

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Members of the Council for 1922

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Chas. T. Main ..... Boston, Mass.
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Edwin S. Carman ..... Cleveland, O.

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William H. Wiley ..... New York, N. Y.

SECRETARY

Calvin W. Rice ..... New York, N. Y.

[1] A. L. Rice elected to fill unexpired term of Mr. Strothman, deceased, May 8, 1922.

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STANDING ADMINISTRATIVE COMMITTEES

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Note: — Numbers in parentheses indicate the number of years the member has yet to serve.

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A. S. M. E. POWER TEST CODES

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*N. A. Carle
E. R. Fish
J. A. Seymour
*W. F. Uhl

Terms expire September 30, 1923

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G. A. Goodenough
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E. N. Trump
*A. C. Wood

Terms expire September 30, 1924

F. Hodgkinson
*D. S. Jacobus
L. F. Moody
H. F. Smith
L. E. Strothman

Terms expire September 30, 1925

*F. R. Low
*L. P. Breckenridge
*R. H. Fernald
C. F. Hirshfeld
R. J. S. Pigott

Terms expire September 30, 1926

A. G. Christie Paul Diserens C. E. Lucke E. F. Miller G. A. Orrok

**Members of the Executive Committee*

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REGULAR NOMINATING COMMITTEE 1922

Nominated by Local Sections Delegates and Elected by Membership Assembled at Annual Meeting

Group	*Local Sections*	*Committee*	*Alternates*
1	Boston, Worcester, Providence, Connecticut	A. W. Honywill, Jr. *Hartford*	H. E. Reed *Worcester*
2	New York (Metropolitan Section).	H. H. Barnes *New York*	G. I. Rhodes *New York*
3	Philadelphia, Lehigh Valley, Baltimore, Washington, D. C., Virginia, Atlanta, Birmingham, Central Pennsylvania, Plainfield.	W. W. Varney, *Baltimore*	H. W. Bussey *Atlanta, Ga.*

4	Eastern New York, Utica, Rochester, Syracuse, Buffalo, Ontario, Erie, Akron, Cleveland, Toledo, Columbus, Pittsburgh, Utica	C. W. Bennett *Pittsburgh*	R. W. Angus *Toronto, Can.*
5	Chicago, Detroit, Milwaukee, Minneapolis, St. Paul, Indianapolis, Cincinnati, Tri-Cities	Paul Doty *St. Paul, Minn.*	W. A. Hanley *Indianapolis*
6	St. Louis, New Orleans, Houston, Mid-Continent, Colorado, Kansas City	V. J. Azbe *St. Louis*	L. D. Crain *Denver*
7	Western Washington, Inland Empire, Oregon, Los Angeles, San Francisco	F. S. King *Hollywood*	R. L. Dougherty *Los Angeles*

REPRESENTATIVES ON JOINT ACTIVITIES

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE, SECTION ON ENGINEERING

Ira N. Hollis — Dugald C. Jackson

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A. S. M. E. Representatives

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CODE OF ETHICS

Special Committee

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H. J. O. Hinchey — Robert Sibley
J. V. Martenis — C. T. Main

FEDERATED AMERICAN ENGINEERING SOCIETIES

*A. S. M. E. Delegates on American Engineering Council**

Terms expiring Dec. 31, 1922	*Terms expiring Dec. 31, 1923*
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L. P. Alford	Chas. A. Booth
M. E. Cooley	Gano Dunn
A. M. Greene, Jr.	H. H. Esselstyn
D. S. Kimball	W. S. Lee
A. L. Rice	I. E. Moultrop
Alternates	J. A. Stevens
Chas. Whiting Baker	A. E. Walden
Fred J. Miller	P. F. Walker

*The Governing Board of The Federated American Engineering Societies.

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D. S. Jacobus
Term expires Feb., 1924

J. H. Barr
Term expires Feb., 1925

JOHN FRITZ MEDAL BOARD OF AWARD

A. S. M. E.

Ambrose Swasey, *Chairman*
Term expires Feb., 1926

Henry B. Sargent
Term expires Feb., 1924

Fred J. Miller
Term expires Feb., 1923

Walter M. McFarland
Term expires Feb., 1925

JOSEPH A. HOLMES MEMORIAL BOARD

William H. Bixby

NATIONAL RESEARCH COUNCIL, ENGINEERING DIVISION

Arthur M. Greene, Jr.
Term expires June 30, 1922

John H. Barr
Term expires June 30, 1923

D. S. Jacobus
Term expires June 30, 1924

UNITED ENGINEERING SOCIETY BOARD OF TRUSTEES

Geo. M. Basford
Term expires Jan., 1923

Worcester R. Warner
Term expires Jan., 1924

W. L. Saunders
Term expires Jan., 1925

WESTERN SOCIETY OF ENGINEERS, WASHINGTON AWARD

Charles F. Brush
Term expires Feb., 1923

Chas. Whiting Baker
Term expires Feb., 1922

SUMMARY OF MEMBERSHIP BY RESIDENCE

United States

Alabama	110	Nebraska	30
Alaska	2	Nevada	5
Arizona	30	New Hampshire	60
Arkansas	16	New Jersey	1396
California	610	New Mexico	15
Canal Zone	8	New York	3682
Colorado	80	North Carolina	75
Connecticut	770	North Dakota	10
Delaware	90	Ohio	1180
District of Columbia	200	Oklahoma	125
Florida	50	Oregon	50
Georgia	150	Pennsylvania	1734
Hawaiian Islands	26	Philippine Islands	37
Idaho	10	Porto Rico	15
Illinois	1140	Rhode Island	180
Indiana	270	South Carolina	40
Iowa	55	South Dakota	10
Kansas	60	Tennessee	105
Kentucky	55	Texas	149
Louisiana	115	Utah	20
Maine	45	Vermont	40
Maryland	215	Virginia	126
Massachusetts	1400	Washington	113
Michigan	580	West Virginia	62
Minnesota	140	Wisconsin	339
Mississippi	15	Wyoming	12
Missouri	280		
Montana	25	Total	16317

Foreign Countries

Africa	15	Italy	10
Asia		Norway	10
China	27	Roumania	2
Dutch East Indies	5	Russia	5
India	40	Spain	7
Japan	37	Sweden	15
Singapore Straits Settlements	3	Switzerland	10
Australia	23	Turkey	3
Central America	5	North America	
Europe		Canada	260
Belgium	7	Mexico	46
Czechoslovakia	3	South America	82
Denmark	5	West Indies	
Finland	5	Cuba	65
France	44	Jamaica	1
Germany	30	Santo Domingo	12
Great Britain	110	Trinidad	2
Greece	1		
Holland	3	Total in Foreign Countries	893

Summary of Membership by Residence

Membership in United States	16157
Membership in Foreign Countries	893
Present address unknown	160
Total Membership	17210

Summary of Membership by Grades

Honorary Members	17
Members	8148
Associates	861
Associate-Members	4107
Juniors	4077
Total	17210

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 44 — 1922

THE activities of the Society for 1922 are recorded in Volume 44. In this volume will be found papers and discussions presented at the Spring Meeting held at Atlanta and at the Annual Meeting held in New York. In addition are other papers which the Committee has deemed of sufficient interest to include in Transactions.

In selecting material for Transactions the Committee is guided by the principle of choosing material of permanent and technical value. For this reason many discussions are condensed and some papers and addresses of less technical interest are given in abstract form. In most cases more complete forms of the papers may be found in MECHANICAL ENGINEERING.

DEXTER S. KIMBALL

Dexter S. Kimball was born in New River, New Brunswick, Canada, in 1865. He was graduated from Leland Stanford, Jr. University in 1896 with the degree of A.B. He received his M.E. degree in 1913.

He served his apprenticeship with Pope & Talbot, Port Gamble, Wash., and was employed for ten years in the shops and the engineering department of the Union Iron Works, San Francisco. In 1898 he became designing engineer for the Anaconda Mining Co. For three years he served as assistant professor of machine design at Sibley College, Cornell University, when he became works manager of the Stanley Electric Manufacturing Co., Pittsfield,

Mass. In 1904 he returned to Cornell University as professor of machine design and construction, and subsequently occupied the joint chair of machine design and construction and industrial engineering. He became dean of the College of Engineering in 1920. During the war he was for some time Acting President of Cornell University and fuel administrator for Tompkins County during the entire period.

Since 1911 Dean Kimball has been a member of the Council on Industrial Engineering, New York State Department of Education. He is also a member of the Society for the Promotion of Engineering Education, the Society of Industrial Engineers, and of the honorary societies Sigma Xi and Tau Beta Pi. He is vice-president of The Federated American Engineering Societies, vice-president of the Society for the Promotion of Engineering Education and past-president of Tau Beta Pi. He was appointed by President Harding to the Board of Visitors of the U. S. Naval Academy for the year 1922.

He is co-author with John H. Barr of Elements of Machine Design and author of Industrial Education, Principles of Industrial Organization, Elements of Cost Finding, and Plant Management, as well as of many contributions to the technical press.

Dean Kimball became a member of the Society in 1900. He has served as chairman of the Committee on Meetings and Program of the Society, having charge of the professional features of the Annual and Spring Meetings. He has also served on the Committee on Aims and Organization and as chairman of the subcommittee on Relation of the Engineer to his Work. He was elected manager in 1919.

No. 1838

SOCIETY AFFAIRS

THE SPRING MEETING

Atlanta, Ga., May 8 to 11

THE 1922 Spring Meeting was the first to be held in the South since the one at New Orleans in 1916. Because of local interest in these subjects, textiles and hydroelectric power development were stressed at the meeting. The local committee took advantage of Atlanta's geographical situation to arrange for excursions to Greenville, S. C., for an inspection trip to textile mills and bleacheries on Friday, May 12, to the Duncan Mills, the Branden Mills, the Judson Mills, the American Spinning Co., the Woodside Mills, and the Union Bleachery and Finishing Co. An excursion to Birmingham, Ala., on Friday, May 12, was arranged for the inspection of the American Cast Iron Pipe Co., and of the Ensley and the Fairfield Plants of the Tennessee Coal, Iron and Railroad Co. On Saturday, May 13, an excursion was arranged to Muscle Shoals, Ala., for the inspection of the nitrate plants and the Wilson Dam.

Previous to the meeting, under the auspices of the Virginia Section, a meeting was held at Charlottesville, Va., on May 5 and 6 at the University of Virginia. At the morning session Charles H. Fish of the Virginia Chapter of the Institute of Architects delivered an address on How to Organize an Engineering Problem. In the afternoon there was held a joint session of this Society and the Society for the Promotion of Engineering Education. The subject of this meeting was How Can the Industries Aid Engineering Education. In the evening Dean Dexter S. Kimball, President of the Society, delivered an address on The Contents of a Liberal Education. On Saturday, May 6, there was an automobile trip through the environs of Charlottesville with a visit to Monticello and a tour of the grounds and laboratories of the University of Virginia.

Registration for the Spring Meeting at Atlanta opened at the Piedmont Hotel, 8:30, Monday, May 8. A meeting of the Council, a conference of local sections' delegates and a meeting of the Nominating Committee constituted the business of the morning while the afternoon was devoted to a business meeting. At the business meeting the proposed new Constitution was presented for its first reading and with a few slight amendments was ordered submitted to letter ballot of the membership sixty days before the Annual Meeting of the Society in December. At the conclusion of the discussion on the Constitution Professor Christie, Chairman of the Committee on the Joint Code of Ethics, made a report for his Committee. It was moved that the revision of the Code be referred to members for letter ballot.

On Tuesday morning simultaneous sessions were held on textile machinery and on materials handling. There was also a general session and a public hearing upon the Power Test Codes. On Wednesday morning there was a second session on textile machinery held simultaneously with a second general session and a session on fuels under the auspices of the Fuels Division. On Thursday morning there were three sessions on management and power under the auspices of these divisions respectively and a symposium on welding under the auspices of the A.S.M.E. Boiler Code Committee and the American Welding Society.

Tuesday afternoon was devoted to local excursions to points of interest around Atlanta. Wednesday afternoon was devoted to a barbecue at the East Lake Country Club which was followed by an eighteen-hole golf match and an exhibition of boxing. On Thursday afternoon there was a student branch conference and a theatre party at the Howard Theatre.

Throughout the meeting a number of Atlanta ladies were present at the headquarters hotel, and the visiting ladies were accorded every courtesy. Impromptu automobile trips were organized and social gatherings arranged during the mornings when the technical sessions were in progress. The crowning social event for the ladies was the serving of tea at the Piedmont Driving Club on Tuesday.

The assistance rendered by the students of the Georgia School of Technology is worth special mention. Throughout the meeting they gave dependable service at the Information Bureau, at the Registration Desk, in the various railroad stations and in the actual conduct of the meeting.

The general local committee in charge of the Atlanta meeting consisted of H. E. Bussey, Chairman; R. S. King, Vice-Chairman; J. T. Wikle, Secretary; E. A. Brooks, Robert Gregg, N. C. Harrison, H. E. Lindley, Thomas M. Nalon, and Earl F. Scott. Chairmen of the sub-committees were as follows: Earl F. Scott, Finance; N. C. Harrison, Entertainment; Mrs. Earl F. Scott, Ladies Social; W. H. Holby, Transportation; W. J. Neville, Hotels; R. S. King, Information; Earl F. Scott, Publicity; Earl F. Scott, Reception.

PROGRAM

Monday Morning, May 8

Opening of Headquarters and Registration Bureau.
Council Meeting.
Conference of Local Sections' Delegates.
Meeting of the Regular Nominating Committee.

Monday Afternoon

BUSINESS MEETING

Announcement of result of ballot on amendments to the Constitution; first reading of proposed new Constitution; report of Special Committee on Code of Ethics; report on American Engineering Council.

Monday Evening

Informal Dance.

Tuesday Morning, May 9

SIMULTANEOUS SESSIONS

TEXTILE MACHINERY (I)

Cotton-Ginning Machinery, S. E. Gillespie.

MATERIALS HANDLING

Material Handling Equipment as Used in the Iron and Steel Industry, F. L. Leach.

GENERAL (I)

The Accuracy of Boiler Tests, Alfred Cotton.

Using Exhaust Energy in Reciprocating Engines, J. Stumpf and C. C. Trump.

Public hearing on Power Test Codes.

Tuesday Afternoon

Local Excursions.

Tuesday Evening

Dinner Dance.

Wednesday Morning, May 10

SIMULTANEOUS SESSIONS

TEXTILE MACHINERY (II)

Weaving Machinery, L. B. Jenckes.[1]

Extraction of Oil from Vegetable Matter, J. Davidson.[2]

Modern Shop Practice in the Building of Revolving Flat Cards, F. E. Banfield, Jr.[2]

FUELS

Reduction of Fuel Wastes in the Steel Industry, F. G. Cutler.

Boiler-Room Performance and Practice at Colfax Station, Duquesne Light Co., C. W. E. Clarke.

The Control of Boiler Operation, E. A. Uehling.

GENERAL (II)

Centrifugal Castings, Leon Cammen.

The Muscle Shoals Plant and the Nitrogen Supply, Maj. J. K. Clement.[1]

Heat Losses from Bare and Covered Wrought Iron Pipe at Temperatures up to 800 deg. fahr., R. H. Heilman.

The Evaporation of a Liquid Into a Gas, W. K. Lewis.

Wednesday Afternoon

Barbecue and golf exhibition match at the East Lake Country Club

Thursday Morning, May 11

SIMULTANEOUS SESSIONS

MANAGEMENT

Management Applied to Textile Plants, George S. Harris.[1]

The Southern Worker — His History and Character, F. H. Neeley.[2]

POWER

Power Development in the Southeast, Chas. G. Adsit.

Economics of Water-Power Development, C. A. Mees.[2]

Hydroelectric Power-Plant Design, John A. Sirnit.

Report Upon Efficiency Tests of a 60,000-kw., Cross-Compound, Triple-Cylinder Steam Turbine, H. B. Reynolds and W. F. Hovey.

[1] Published in Mechanical Engineering, June 1922.
[2] Published in Mechanical Engineering, May 1922.
[3] Published in Mechanical Engineering, July 1922.

SYMPOSIUM ON WELDING

Strength of Electrically Welded Pressure Containers, R. J. Roark.[1]

Some Principles of the Construction of Unfired Pressure Vessels, S. W. Miller.[2]

Steel for Forge Welding, F. N. Speller.

Thursday Afternoon

Meeting of regular Nominating Committee.
Student Branch Conference.
Theatre Party.

THE ANNUAL MEETING

New York, N. Y., December 4 to 7

The forty-third Annual Meeting was notable for its well-balanced, diversified program with papers and discussion. The four days of the meeting, December 4 through 7, 1922, formed a very intense period, however, for during that time there were twenty-four sessions, fifty-three committee meetings, and four social events. This was followed by the National Exposition of Power and Mechanical Engineering which opened on Thursday, December 7, at one o'clock at the Grand Central Palace. It was a remarkable week and the 1836 who registered left the meeting with a feeling of having attended a very successful affair.

The presidential address of Dean Dexter S. Kimball attracted wide attention and gave mighty inspiration in its challenge to the engineer to assume his proper responsibility in maintaining and developing civilization. He was given an ovation at its presentation on Monday evening of the meeting.

Perhaps the most interesting session of the meeting was that on Engineering and Economics held on Wednesday evening, December 6. President Dexter S. Kimball presided over a program arranged jointly by the A.S.M.E. Management Division and the American Economic Association. Papers were presented by Dr. W. C. Mitchell on Making Goods and Making Money, and by E. M. Herr on The Human Problem in Industry; and H. R. Seager, E. F. DuBrul, F. J. Miller, and J. L. Harrington contributed to the discussion.

[1] Published in Mechanical Engineering, April 1922.
[2] Published in Mechanical Engineering, June 1922.

The Council met on Monday to transact routine business, and on Friday to install the new officers and consider questions arising from the proceedings of the business meeting. John Lyle Harrington was formally introduced as President; as were the new Vice-Presidents, W. S. Finlay, Jr., W. H. Kenerson, E. F. Scott, and H. H. Vaughan; and Managers, A. G. Christie, J. H. Herron and R. V. Wright.

This Annual Meeting was also the scene of a very remarkable conference of Local Sections Delegates which lasted throughout the week. On Monday the delegates met all day to talk over Local Sections problems, breaking the gathering, however, for luncheon with the Council. During the rest of the week the Local Sections Delegates met with the various Standing Committees of the Society for the discussion of national Society problems, and on Friday they attended the final meeting of the Local Sections Delegates.

On Monday afternoon, December 6, the annual business session [1] was held. Its feature was the presentation of the report of Calvin W. Rice, Secretary of the Society, on his recent trip to South America.[2] Mr. Rice preceded this report by his customary summary of the work of the Standing Committees of the Society. The formal report of the Tellers on the ballot for the new Constitution and Code of Ethics was read.

Life membership in the Society was also presented to Prof. R. C. H. Heck for a paper on Steam Formulas [3] which was adjudged by the Committee on Awards and Prizes as the best one appearing in the 1920 volume of Transactions. The Junior prizes were awarded to R. H. Heilman of Pittsburgh for his paper [4] on Heat Losses from Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Deg. Fahr., and to F. L. Kallam of Los Angeles, Cal., for his paper on The Investigation of the Thermal Conductivity of Liquids.

E. S. Carman, Junior Past-President and Chairman of the A.S.M.E. delegation to the American Engineering Council, emphasized the fact that the organization of national, state, and local societies composing The Federated American Engineering Societies is concerned more particularly with those engineering activities that are of great civic import.

[1] See Mechanical Engineering, January 1923, p. 59.

[2] See Mechanical Engineering, January 1923, p. 72.

[3] See Trans. vol. 42, p. 711.

[4] Published in this volume of Transactions.

L. W. Wallace, Executive Secretary of the Federation, gave a convincing report of the activities of the Federation.

Following Mr. Wallace, Rudolph P. Miller gave a report on the National Board of Jurisdiction Awards in the Building Industry.[1]

On Wednesday afternoon, December 6, Dr. William H. Kenerson presided over a gathering of representatives of 36 Student Branches at which a number of problems relating to the activities of the Branches and their relation to the Society were given careful consideration.

The meeting furnished its usual quota of valuable opportunities for good-fellowship. Although the President's Reception, usually held on Tuesday evening, was held this year on Monday, it was exceedingly well attended. The informal dinner and smoker on Tuesday evening gave the desired opportunity for the men to get together, and on Wednesday afternoon the ladies served tea. The dinner dance Thursday night was a success from every point of view.

Throughout the week a ladies' committee was in attendance at the building and large groups of visitors were taken to the Museum of Natural History, to a private gallery of modern American paintings, to inspect the Pictorial Review building, the Hecksher Foundation and the Museum of the American Indian, and to a Fashion Show at Wanamaker's where tea was served.

The more technical excursions included visits to the Hell Gate Station of the United Electric Light and Power Company, the Broadcasting Station WEAF of the American Telephone & Telegraph Company, and the plant of the Wheeler Condenser Company at Carteret, N. J. Large groups visited the McGraw-Hill building for luncheon on Thursday, and the Battleship *Maryland* at the Brooklyn Navy Yard on Friday. A very interesting demonstration of Christie's amphibious gun mount was held on Tuesday afternoon, when this new device traveled over smooth roads on wheels, over broken country on its caterpillar tread, and then propelled itself across the Hudson.

[1] See MECHANICAL ENGINEERING, January 1923, p. 75.

PROGRAM

Monday Morning, December 4

Conference of Local Sections' Delegates.
Council Meeting.
Boiler Code: Public Hearing on Power-Boiler Section.

Monday Afternoon

Joint Session with American Society of Refrigerating Engineers.
Design of Cooling Towers, C. S. Robinson.
Economic Thickness of Insulation in Refrigerating Field, Percy Nicholls.

Monday Evening

Presidential Address, Dexter S. Kimball.
Report of Tellers of Election.
Introduction of President-Elect Harrington.
Reception.

Tuesday Morning, December 5

SIMULTANEOUS SESSIONS

MANAGEMENT

Ten Years' Progress in Management, L. P. Alford.
Relieving Industry of Burden, Wallace Clark.[1]
Progress Reports of Committees on Standardization of Terminology and Standardization of Graphics.

MATERIALS HANDLING

Ash Handling, John Hunter and Alfred Cotton.

MACHINE SHOP

A New System of Helical Involute Gearing for Use on Metal Planers, F. E. Cardullo.
Spherical Gears, C. H. Logue.
Testing Involute Spur Gears, M. Estabrook.[2]
Application of Hydraulic Transmission Variable Speed Drive to Machine Tools and Manufacturing Processes, Walter Ferris.
Power Required for Removing Metal, F. A. Parsons.

PUBLIC HEARINGS

Power Test Codes: Feedwater Heaters, Reciprocating Displacement Pumps.
Boiler Code: Heating Boiler Section.

[1] Published in Mechanical Engineering, December 1922.
[2] Published in Mechanical Engineering, January 1923.

Tuesday Afternoon

SIMULTANEOUS SESSIONS

EDUCATION AND TRAINING FOR THE INDUSTRIES

REPORT OF COMMITTEE, W. W. Nichols, *Chairman.*

REPORTS ON: EXTENSION AND CORRESPONDENCE SCHOOLS, James A. Moyer.

INDUSTRIAL EDUCATION AS REPRESENTED IN SCHOOLS, C. R. Richards.

SCHOOLS FOR APPRENTICES AND SHOP TRAINING, R. L. Sackett.

PUBLIC HEARING

BOILER CODE: Heater Boiler Section.

RESEARCH

A NEW METHOD OF DETERMINING THE EFFECT OF SPEED UPON THE STRENGTH OF GEAR TEETH, Wilfred Lewis.[1]

EFFECT OF PULSATIONS ON FLOW OF GASES, H. Judd and D. B. Pheley.

ORIFICE COEFFICIENTS — DATA AND RESULTS OF TESTS, J. M. Spitzglass.

TORSION OF CRANKSHAFTS, S. Timoshenko.

EFFECTS OF LARGE SUDDEN STEAM DISCHARGES FROM BOILERS — Report of Research Committee on Sudden Initial Pop Lift of Safety Valves.

GENERAL I

REFINERY AND ROLLING MILL FOR MONEL METAL, HUNTINGTON, W. VA., W. L. Wotherspoon.

THE SIZE OF A DRY-VACUUM PUMP TO EMPLOY IN A GIVEN CASE, E. W. Noyes and H. V. Sturtevant.

DIESEL ENGINE CLUTCH USED IN GERMAN SUBMARINE *U-117*, William H. Nicholson.

STEAM TABLES

DISCUSSION OF PROGRESS IN RESEARCHES.

Tuesday Evening

SMOKER AND DINNER.

Wednesday Morning, December 6

SIMULTANEOUS SESSIONS

FUELS

SYMPOSIUM ON STOKERS:

DEVELOPMENT AND USE OF THE MODERN CHAIN GRATE, T. A. Marsh.

OVERFEED STOKERS OF THE INCLINED TYPE, G. I. Bouton.

DESIGN AND OPERATION OF UNDERFEED STOKERS, H. F. Lawrence.

CHRONOLOGICAL HISTORY OF STOKER DEVELOPMENT TO THE PRESENT DAY, A. H. Blackburn.

[1] Published in MECHANICAL ENGINEERING, December 1922.

RAILROAD

Steam Distribution in the Locomotive, G. H. Hartman.[1]
Mechanical Draft for Locomotives, F. H. C. Coppus.
Stresses in Locomotive Frames, R. Eksergian.

GENERAL II

Design of Flywheels for Motor-Driven Impulse Machines, C. O. Rhys.

Stresses in Cylindrical Shaped Rotors of Uniform Diameter, C. M. Laffoon.

Stress Distribution in Electric-Railway Motor Pinions as Determined by the Photo-Elastic Method, Paul Heymans and A. L. Kimball, Jr.

Wednesday Afternoon

Business Meeting.
Student Session.
Ladies' Tea and Reception.

Wednesday Evening

ENGINEERING AND ECONOMICS

Making Goods and Making Money, Wesley C. Mitchell.[2]
The Human Problem in Industry, E. M. Herr.[2]

Thursday Morning, December 7

SIMULTANEOUS SESSIONS

POWER

Tests of a Type W Stirling Boiler at the Connors Creek Power House of the Detroit Edison Company, Paul W. Thompson.

Feed Heating for High Thermal Efficiency, Linn Helander.

High-Temperature and High-Pressure Steam Lines, B. N. Broido.

The Elasticity of Pipe Bends, Sabin Crocker and S. S. Sanford.

The Commercial Economy of High Pressure and High Superheat in the Central Station, Geo. A. Orrok and W. S. Morrison.

SAFETY ENGINEERING

Safety Codes, M. G. Lloyd.

Safety Engineering in Connection with the Compression of Gases, A. D. Risteen.

Safety Code for Grinding Wheels, G. E. Sanford.

Some Hazards of the Logging Industry, John A. Dickinson.

[1] Published in Mechanical Engineering, December 1922.
[2] Published in Mechanical Engineering, January 1923.

STANDARDIZATION

Program for the Standardization of Paper and Printing Machinery, Wm. J. Eynon.

Size Standardization by Preferred Numbers, C. F. Hirshfeld and C. H. Berry.[1]

Thursday Afternoon

SIMULTANEOUS SESSIONS

ORDNANCE

Motion Pictures: Some Recent Developments of Mobile Ordnance

Machining and Lapping Very Deep Holes, J. B. Rose.

Methods Used in Manufacture of Gun Mechanism, R. A. Vail.

AERONAUTICS

Influence of Design on Cost of Operating Airplanes, A. Black.[2]

The Airship for Long Haul Heavy Traffic Service, R. H. Upson.

Night Flying in Airways, H. R. Harris and D. L. Brunner.

Air Navigation, R. W. Wilson and M. D. Hersey.

FOREST PRODUCTS

New Factors Which Are Influencing Woodworking Machinery Design, S. Madsen.

Control of Lumber Cutting Waste and Production, C. M. Bigelow.[1]

Some Engineering Aspects of the Design of Musical Instruments, W. B. White.

Lumber Dry Kilns, Thos. D. Perry.

Lumber Standardization, F. F. Murray.

Thursday Evening

Dinner Dance.

[1] Published in Mechanical Engineering, January 1923.

[2] Published in Mechanical Engineering, December 1922.

No. 1839

COTTON-GINNING MACHINERY

By Solomon E. Gillespie, Dallas, Tex.
Member of the Society

Of the two classes of cotton grown in the United States — sea island or long-staple, and upland or green-seed — the former does not adhere to the seed and the lint can be pulled off clean by running it through rollers set close together, leaving the seed behind. Upland cotton, on the other hand, adheres to the seed, and the present paper is devoted to a description of the apparatus developed since Eli Whitney's epoch-making invention in 1792 for mechanically handling the raw seed cotton, removing the lint therefrom, and forming it into shape suitable for delivery to the baling press.

The cotton-ginning process comprises (a) elevating the cotton from the wagons in which it is brought to the ginnery to (b) a cleaner where the dirt, sand, leaf trash and other foreign substances are extracted; (c) elevating the cleaned cotton to (d) feeders which deliver it uniformly to (e) the gins where the lint is removed from the seed by the teeth of circular saws mounted at close intervals on a rotating shaft, from which teeth it is (f) cleaned off pneumatically and (g) carried to a condenser, where it is formed into a "bat" and delivered to a press box for packing, wrapping and pressing. The latest developments of the various devices employed in these steps are described and illustrated in the paper, and the author has included much valuable information for designers in the way of dimensions, relative proportions, speeds of operation, etc.

LONG before the dawn of history the cotton plant was cultivated in various parts of the world, and the earliest records of spinning its fleecy bolls are of such antiquity that it is difficult to obtain satisfactory evidence of its beginning. The name "cotton" itself is of Oriental origin, being derived from the Arabic "koton" or "gootn."

2 India is known as the motherland of cotton, and its earliest history is found in the Rig Veda hymn, composed fifteen centuries before the Christian era, which honors the "threads of the loom," and also three thousand years ago it was used as an ornamental shrub in Egypt.

Presented at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

3 Alexander the Great acquainted Europe with India, and with that singular plant "vegetable wool," and often referred to "the trees of India bearing wool." Cotton was first imported into England in 1298 A.D., and the manufacture of cotton goods thereafter became one of its greatest industries. In the United States cotton became an early experimental plant, beginning in Virginia in 1600 A.D. and after that date its growth was very rapid. During the war of the American Revolution and immediately thereafter, cotton culture began to receive considerable attention in the Southern States; along the coast, the valuable sea island cotton was raised, and later, in the interior, the upland or green-seed cotton was raised.

4 Before taking up the ginning of cotton it is well to consider its principal characteristics. There are two general classes or distinctive kinds of cotton grown in the United States, "sea island" and "upland." Sea island cotton does not adhere to the seed, and the lint can easily be pulled off clean, leaving the seed perfectly smooth. The staple varies from $1\frac{1}{2}$ in. to $2\frac{1}{4}$ in. in length, and is of a light creamy silk color. While strong, it is finer than other kinds, and being about 0.00063 in. in diameter, has a more beautiful luster.

5 Upland cotton adheres to the seed, and appears to grow out of them more like wool from a sheep's back. The seed, after being divested of lint as well as possible, still have a woolly appearance, and in a great many varieties a greenish color and hence upland cotton is referred to as "green-seed" cotton. The staple varies in length from $\frac{5}{8}$ in. to $1\frac{1}{8}$ in. and its color ranges from white to a creamy tint. The staple is inferior in strength to the sea island type and coarser, being 0.00076 in. in diameter. The fibers are soft, elastic, moist and pliable, but the natural twist is rather inferior in character to that of sea island cotton, being irregular.

6 Good commercial cotton must possess certain well-defined external and internal characteristics. The external qualities, which are apparent to the touch or visible to the naked eye are: (*a*) Length of fiber, (*b*) fineness or smallness of diameter, (*c*) evenness and smoothness, (*d*) elasticity, (*e*) tensile strength, (*f*) color. The internal characteristics, which are discernible by a microscope, are: (*a*) Hollowness, or tubular construction, (*b*) natural twist, due to the collapsed tube, (*c*) corrugated edges, (*d*) moisture.

7 These characteristics are necessary to constitute a good mature cotton fiber, are essential in effecting the close union of the

filaments, and make cotton superior to any other fiber, vegetable or animal. And since the cotton fiber has such a delicate structure and great value, it is necessary that the machinery used in its ginning and cleaning be of such character that the least possible damage will be done to the staple in the process.

8 The first method to be employed in separating cotton lint from the seed was that of hand picking. The next method, originated in India about 300 B.C., made use of rollers, which when running close together would pull the lint through and leave the seed behind. The roller gin is now used principally for ginning sea island or long-staple cotton and has gone through a long process of development and improvement.

9 In the United States the great problem that presented itself in the early days of the industry was that of removing the seed from the upland or green-seed cotton. The roller gin in its crudely developed state at that time would handle the sea island cotton to a certain extent, but was not adaptable to and suitable for the upland cotton.

10 In 1792 Eli Whitney, of Massachusetts, went by boat to Savannah, Georgia, and there developed the first cotton gin, for which a patent was granted him two years later. The original Whitney gin was a hand-power and hand-fed gin, consisting of a horizontal wooden cylinder about 7½ in. in diameter and 2 ft. long, into which wire teeth were driven in rows spaced apart to admit the seed cotton, but with the teeth in each row so close together as not to admit seed. The teeth were all inclined the same way, making an angle of 55 deg. to 60 deg. with a tangent at the point of entry. The cylinder had suitable bearings at each end and a hand crank, and above and in front of the cylinder was a hopper to receive the seed cotton, while directly over it was a breastwork or stripping grate consisting of a timber lined with a metal strip having a series of slots through which the rows of teeth could pass, taking the lint cotton but obstructing the seed. The lint was removed from the teeth by means of a cleaner or brush consisting of four sticks with bristles and so constructed that it formed a tangential cylindrical coacting surface of bristles rotating in an opposite direction to that of the teeth and much faster. It was driven by an endless belt and idler. The gin was not continuous in action, as are the gins of today, but had to have its seed dumped from time to time as they accumulated.

11 Two years later, — May 12, 1796, — a patent was issued

to H. Ogden Holmes of Augusta, Georgia, on a hand-fed and hand-operated gin similar to Whitney's, except that it was continuous in operation, being so constructed that the seed would shed out as fast as they were delinted. This was also the first gin to use metal disks or saws, which were mounted on a square shaft with ends turned down for bearings and the saws separated by space blocks. The brush contained more than four bristle-filled sticks, and was driven in reverse direction to that of the saws and four or five times as fast. This gin would shed the seed continuously as fast as delinted, and the roll of cotton in the roll box would revolve just as it does in the saw gin of today.

12 A little later, larger power-driven gins were employed, Fig. 1, the transmission consisting usually of a large 8-ft. wooden bevel gear mounted on a vertical axis with an extended beam or arm to which a horse or mule was hitched, and a horizontal shaft with a wooden pinion worked in conjunction with the large bevel gear on which shaft a 12-ft. band wheel was mounted and from which the gin was driven at from 200 to 300 r.p.m. by means of a belt. In this system the gin was hand-fed from a platform above. The lint was discharged into a sack suspended beneath a hole in the floor back of the gin, sacking being the first system to be used in handling cotton in the market. Later, however, a lint room was built to receive the lint cotton, which was then carried out of doors in baskets and put into a crude wooden press. This press was of very heavy mill construction, consisting of a single heavy box with discharge doors at the bottom and four large wooden corner struts or tension supports. An elevated platform was built of sufficient height to easily permit the placing of cotton in the press box from above, just below the overhead sills. The press screw was of wood and 15 in. in diameter with levers fixed to the top end and inclined downward to receive a rope to which a horse or mule was attached. Some presses had roofs over them, while others were left out in the open.

13 The ginning system was later improved by the addition of a mechanical feeder over the gin and a single lint condenser back of it, and also by the employment of the steam engine for motive power. As the demand for larger output grew, the ginneries were increased to two and three stands and the presses made more modern by the use of a 5-in. steel screw. In some cases the presses were equipped with large steam cylinders 30 in. to 36 in. in diameter, which operated much more rapidly than the screw press. These

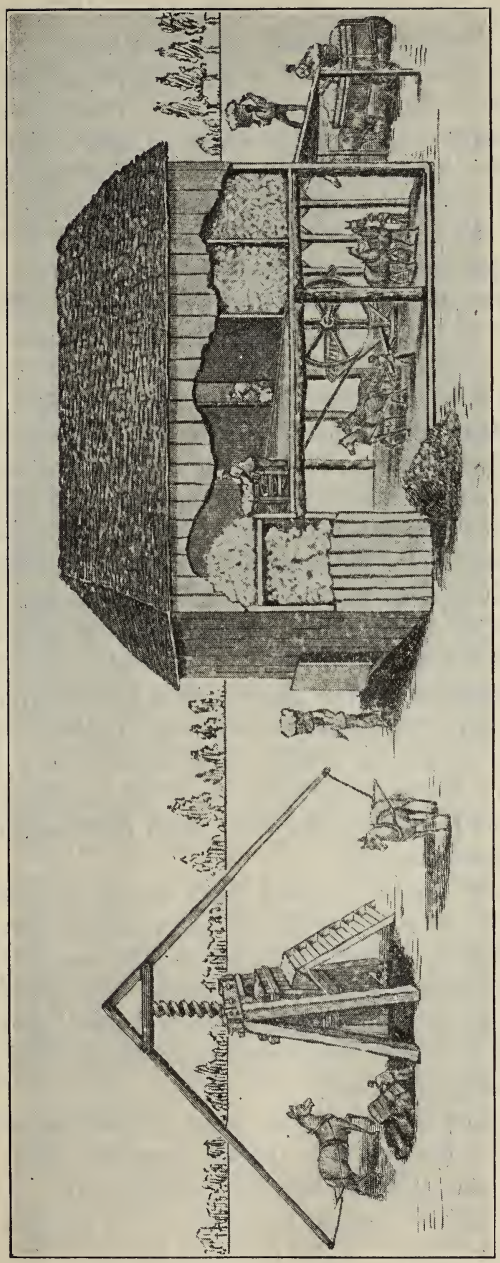

Fig. 1 Early Type of Ginnery

small ginneries were commonly known as plantation gins and were a part of the equipment of each large plantation. As the cotton industry grew, however, and the large plantations were superseded by small ones, custom gins came into use, doing away with the plantation gin and compelling inventors to seek better methods of caring for larger capacities at the gins. At first belt carriers were installed back of each battery of gins, which received the lint from each of the single condensers and delivered it to the press box. This crude system, however, was later replaced by a single lint flue receiving the lint cotton direct from the various gin stands — which were in batteries of from two to six stands — and delivering it to a single large lint condenser or bat former, which then delivered it to the press. The lint flues were first constructed of wood, but later sheet iron was used, the diameter being about 18 in. at the smaller end and increasing as gins were added. There is a story in connection with the development of the lint flue that is related by one who is very high in the cotton-gin business, and who was in close touch with its early history and development. The first flue was a wooden box back of the gins leading to the condenser, and in its construction, it was deemed necessary that a wide belt be in the bottom of the flue to assist in carrying the lint cotton that might fall, and to assist further in its travel; but when time came to start the system the wide belt failed to operate, due to the driving mechanism being out of order, and a surprise resulted in the lint cotton being blown on to the condenser by the air produced by the brushes in the gins, which demonstrated that the wide belt was not necessary.

14 To care further for the large output of cotton at the cotton ginneries, the revolving double-box press was developed. In this press one box acted as a receiving chamber from the condenser and was provided with a tramper or packer which packed 500 lb. of lint cotton in it, after which it was revolved again, bringing the other empty box on to the receiving side while the one filled with cotton was pressed into a bale.

15 The standard cotton-bale boxes for presses are 27 in. by 54 in. by 10 ft.; the cotton is pressed from the latter dimension down to 36 in. and then six ties are put around it. Before being pressed, however, jute bagging is placed at the top and bottom of the cotton for wrappers. Upon releasing the pressure on the bale it usually expands to 40 or 42 in., making a finished bale 27 in. by 54 in. by 40 to 42 in. with a density of 12 lb. per cu. ft. A maximum

pressure of 10,000 lb. is required in packing a bale of cotton in the receiving chamber and 70,000 lb. to press a standard bale.

16 Mechanical elevating systems were also developed for use in the large ginneries. The earlier designs were belt carriers, consisting of slats of wood nailed or riveted to two narrow belts, which would drag the seed cotton along from the point where it was fed on by hand to the point of delivery in the feeders. This process, however, required too much human labor, and one comprising a suction fan with vacuum box and belt distributor was next devised. In this system the cotton was sucked from the wagon through a pipe to the vacuum box, in which there was a wire screen to separate the cotton from the air, the cotton then falling on the spiked-belt distributor below by means of which it was delivered to each gin feeder. The screen area in these vacuum boxes was 1 sq. ft. of ¼-in. mesh screen wire to 14 to 20 saws. The pneumatic elevating system, Fig. 2, was also invented, and consisted of a vacuum chamber or elevated chute over each feeder and gin. This chute consisted of a chamber 12 in. by 53 to 60 in. by 6 ft. high with an air and cotton pipe at the top, a screen box attached on the upper back side and a heavy 12- to 16-oz. canvas valve near the center. This valve was so constructed that when the suction was applied it would close and prevent leakage of air through the valve, causing suction in the telescope pipe in the wagon. This would deliver cotton into the several elevator chutes and then periodically the air would be cut off by means of a valve trip, thus releasing the air suction, and the canvas valves would drop the cotton into the gin feeder. The screen in the pneumatic elevator to separate the cotton from the air is usually light No. 21 gage, ¼-in. mesh wire, and the screen area is about one square foot to 10 saws. The fans used in these two elevating systems are of the steel-plate centrifugal type, varying in size from a 30-in. fan for a two-gin installation to a 45 to 50-in. fan for six gins. The peripheral speed of the suction fan-blast wheels varies from 11,000 ft. per min. to 13,000 ft. per min., which maintains a vacuum of 14 to 16 in. of water.

SEED-COTTON CLEANERS

17 There are many kinds of machines in use for cleaning seed cotton, but the general principle, employed in them all, consists of agitating the cotton on a wire screen of ¼-in. to ⅓-in. mesh, per-

mitting the dirt, sand, leaf trash and other foreign substances to escape. It is customary in some sections of the country, where climatic conditions interfere with the opening of the cotton boll, to provide a boll-breaking attachment to a cleaner. In some sections where labor is scarce, the seed cotton is not carefully picked from the open bolls but is grabbed; that is, cotton, pod and all, are pulled off in a mass. In such cases the cleaning is an important item,

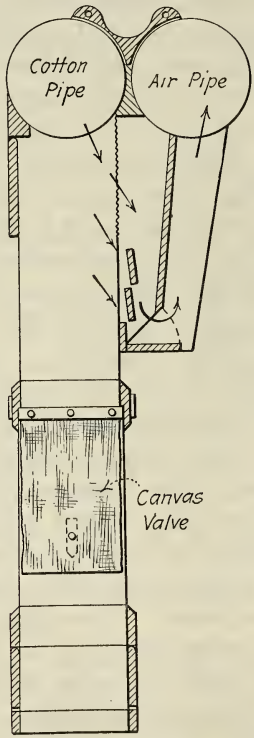

Fig. 2 Pneumatic Elevating System for Large Ginnery

because if not carefully handled the cotton will be damaged in the process. Some cleaning machines therefore have a cylinder of saws to separate the seed cotton from the pod or broken bolls; but this process is best accomplished in the huller gin which will be considered later.

FEEDERS

18 The feeder receives the cotton from the pneumatic elevator or belt distributor above it, and delivers it in a uniform, even

amount to the ginning roll box. There are two general types of feeders. The horizontal slatted-belt type that receives the cotton from the belt distributor or pneumatic elevator on a very slow-moving slatted belt and moves it horizontally up to a spiked drum or picker roller 9 in. to 11 in. in diameter, rotating 180 to 200 r.p.m. and is provided with a series of No. 8 spikes projecting 1 in. from the surface. These spikes lift the cotton over and throw it down a

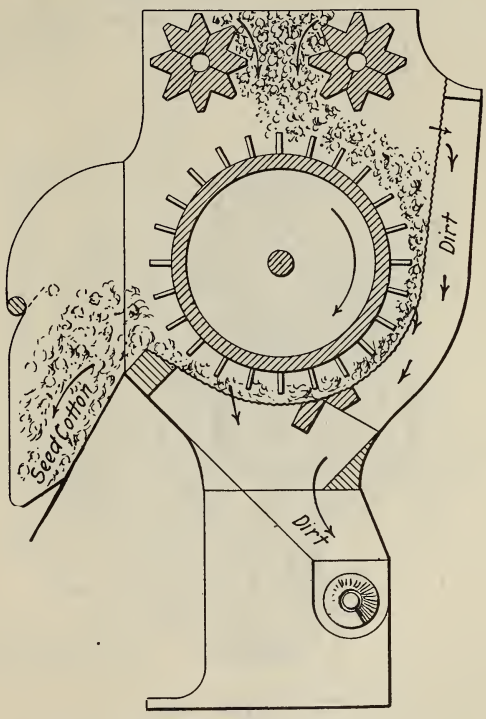

Fig. 3 Feeder for Ginning Roll Box

chute into the ginning-roll box. Another general type, Fig. 3, receives the cotton from the elevator chutes or belt distributor onto two fluted rollers 5 in. to 6 in. in diameter rotating toward each other at $\frac{1}{2}$ to 1 r.p.m. slowly feeding the cotton down where it is taken off by a picker roller similar to the one previously described, and is then whipped around a $\frac{1}{3}$-in. mesh No. 16 gage wire screen that partly surrounds the picker roller at a distance of about $\frac{1}{2}$ in. therefore threshing the light leaf trash and sand out of it before delivering it to the ginning roll box. The latest improved feeders are among the most effective cleaning machines in use.

THE SEED-COTTON GIN

19 The gin stand is the principal machine in the system, as it performs the main work of separating the seed from the lint. There are two general types of saw gins: (*a*) The plain gin, Fig. 4, that gins only clean seed cotton that is substantially free from bolls

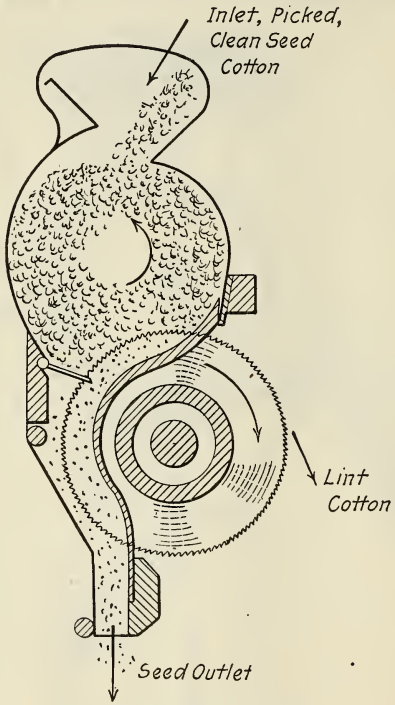

Fig. 4 Plain Gin

and trash and is the original type used; and (*b*) the huller gin, Fig. 5, that separates the seed cotton from the bolls, burrs, leaf trash, dirt and other foreign substances in the front lower roll box, and then carries the seed cotton into the upper roll box where it separates the lint from the seed, thus performing a double ginning operation. The main working parts in both types of gins are the saw cylinder, roll box and a means of removing the lint from the saw teeth after it is separated from the seed.

THE PLAIN GIN

20 The standard saw cylinder is made up of a series of 70 to 80 steel circular saws 10 in. to 12 in. in diameter and 0.035 in. thick, mounted on a $2\frac{3}{16}$-in. to $2\frac{7}{16}$-in. mandrel and separated by wood or iron space blocks 0.6835 in. to 0.7396 in. thick and $4\frac{3}{4}$ in. for 10-in., and 6 in. in diameter for 12-in. saws. The 12-in.

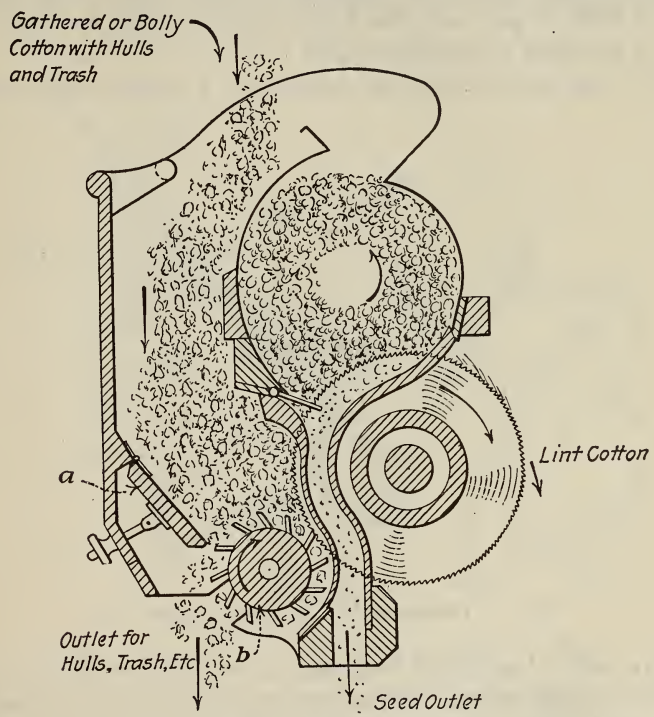

FIG. 5 HULLER GIN

diameter saw is considered best practice. The saw teeth all incline in the direction of rotation. There are seven teeth per inch, which are punched out and then filed smooth. This punching, however, crystallizes the saw steel to a certain extent and often results in defective teeth, with rough edges which are not noticeable to the naked eye but are very distinctly seen with a microscope. Some means should be devised for milling or grinding the teeth instead of punching to insure a perfect smooth tooth free from fracture and crystallization. The usual practice in the design of the tooth is

to make both the front and back edges straight lines. In some cases, however, the front edges have been made straight and the back edges convex resulting in what is known as " hog-back " teeth. The inclination of the teeth edges, Figs. 6 and 7, with a radial line through the point are 36 deg. for the front edge and 57 deg. for the back edge, the two edges making an angle of 21 deg. with each other at the point; this, however, may vary to suit certain working conditions later referred to. The saws rotate with a tooth travel of 1000 to 1250 ft. per. min. for brush gins and 1400 to 1900 ft. per min. for air-blast or pneumatic gins.

21 The roll box, Fig. 8, consists of a series of ribs between

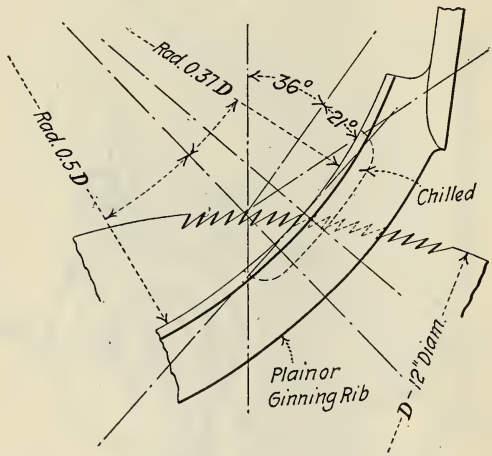

Fig. 6 Detail of Rib and Saw, Plain Gin

the saws with a half-rib at each end, a back hollow above the ribs, an inlet opening or feed inlet at the top, a cover on the front side with a seed grate attached, and a shield or cover for the exposed portion of the saws below the roll box. In the development of the gin it was found by experiment that the process of ginning worked best by placing the roll box above the saws, thus permitting the roll of cotton to rest upon them, Fig. 4; the saw teeth in taking hold of the cotton fibers would cause the whole mass of cotton to revolve in reverse direction, and usually at about a fifth of their velocity, depending upon the quantity of cotton in the roll box and the hardness of the roll.

22 The cotton roll is theoretically round, Fig. 8, and the same diameter as that of the saws; but due to the necessary working

conditions outlined in Pars. 23–26, its perfect roundness is distorted.

23 The saws are caused to project up into the roll box from $1\frac{1}{16}$ in. to $1\frac{5}{16}$ in., $1\frac{3}{16}$ in. being considered the best practice. The cotton roll merely rests upon the saws, and the saw teeth, Fig. 4, only project into it. The cotton roll itself when rotating is hollow due to centrifugal force and the cavity in the center from 25 per cent to 50 per cent of its diameter. It is best practice for

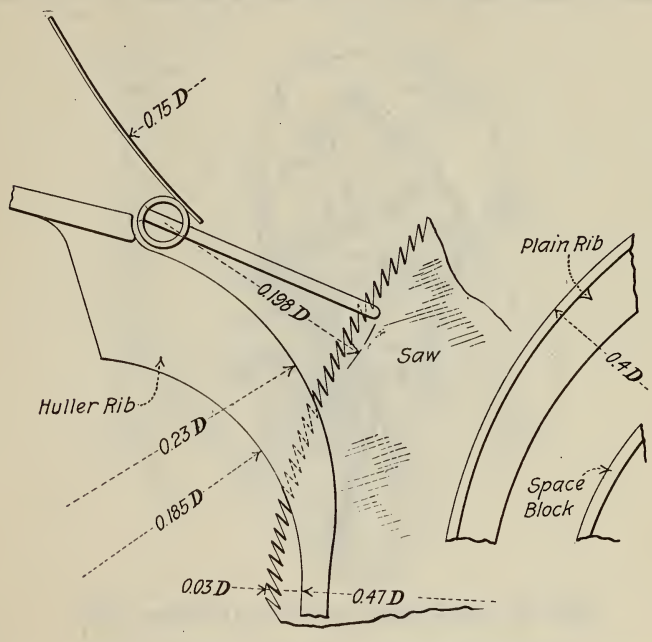

Fig. 7 Detail of Rib and Saw, Huller Gin

the roll of cotton in the roll box to be as soft as possible, that is, with as large a cavity as possible, because this prevents the cotton fibers from wedging into the throats of the saw teeth, cutting and knapping the fibers, and also permits the fibers caught by the saw teeth to disentangle from the mass of the roll without breaking off; this precaution is especially necessary when the saw teeth are traveling at a high velocity.

24 The rib is tangent to the theoretical roll at its lower side; but the curve of the rib cuts up into the theoretical roll with a radius of 37 per cent of the saw diameter D, and at the upper edge of the rib, from $1\frac{1}{2}$ to 2 in. above the point where the saw

teeth pass through there is an offset dropping the working line back tangent to the theoretical roll, to permit the cotton fibers that were pulled between the ribs and not carried away by the saw teeth, Fig. 4, to free themselves from the ribs. The ribs are usually made of cast iron and at the points where the saw teeth pass between them they are chilled for an inch or so each way, Fig. 6, thus making a hard working surface that exhibits very little wear. The space between the ribs and the point where the teeth pass through is 0.125 in. wide and increases in width in each

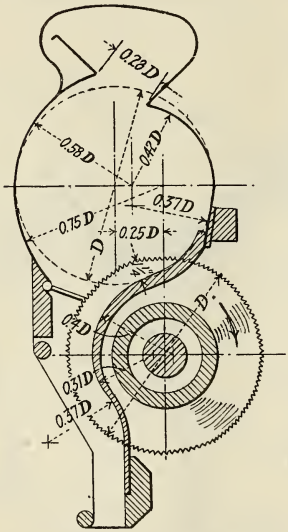

Fig. 8 Proportions of Plain-Gin Roll Box

direction therefrom. The wearing surface of the rib as well as the edges are ground smooth and polished. The lower end of the rib is extended downward beyond the circle of the saw and attached to a rail or bar and the upper end extended just above the offset to attach likewise to a bar or rail, while the inner surface is ground to the curvature (0.37D), of the theoretical roll.

25 The back hollow begins at the upper end of the rib, tangent to the theoretical roll, and curves inward and upward with a radius of 42 per cent of the saw diameter D to give the roll a proper density at the ginning point below and also pack it just before reaching the inlet opening at the top. This opening is 28 per cent of the saw diameter D, and is for the purpose of

admitting cotton into the roll box at the top when operating as a plain gin.

26 The front roll curves are the reverse of those of the rib and back hollow. The front is tangent to the theoretical roll at the intersection of the horizontal center line with it, and curves outward therefrom toward the top, to allow freedom for the admission

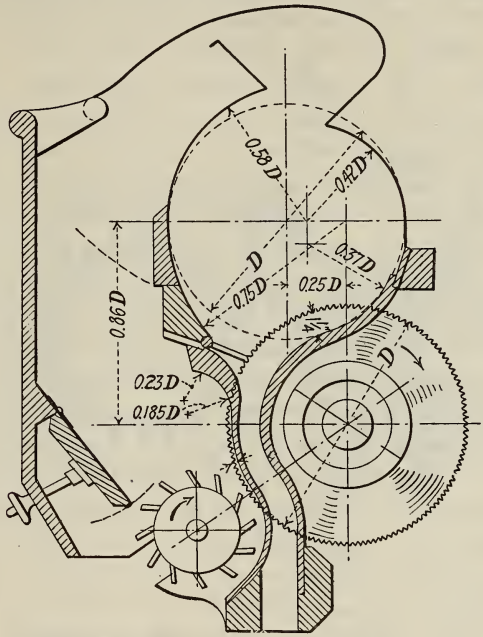

Fig. 9 Proportions of Huller-Gin Roll Box

of cotton, with a radius of 58 per cent of the saw diameter, D. It also curves out from the center line toward the bottom, with a radius of 75 per cent of the saw diameter D, to allow the roll to loosen so that the ginned and cleaned seed will drop out. Near the bottom of this curve of the front there is a seed grate consisting of a pivoted rod with No. 8 fingers projecting out between the saws and extending $\frac{1}{4}$ in. inside the saw-tooth curve. By raising the fingers or grates they press up against the bottom of the cotton roll and retard the shedding of the seed, and by lowering them the seed are shed free; if lowered too much, however, the seed are released before the lint is entirely removed, Fig. 4, the ginning being thus regulated. The seed are shed out in the course of their travel

from the seed grate to the point where the saw teeth pass between the ribs, the seed that are free from lint dropping out of the roll on the ribs below and escaping to the seed conveyor below. The center of the roll box is not directly above the center of the saws because it would cause the ribs below to be too flat and not permit the seed to slide down out of the way, therefore the center roll box is 25 per cent of the diameter D in front of the vertical

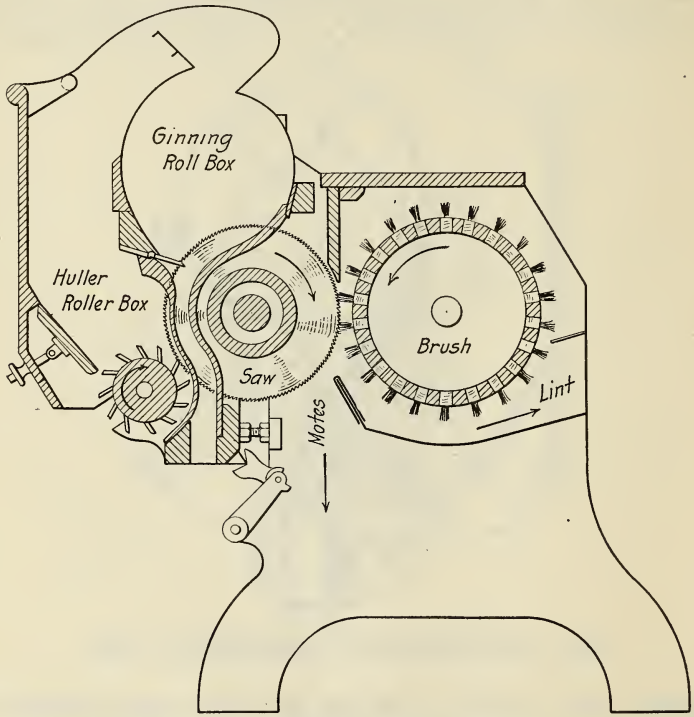

Fig. 10 Brush for Removal of Lint from Saw Teeth of Huller Gin

center line. The exposed portion of the saws below the roll box is covered with a shield to prevent injury to the operator; some roll boxes, however, have fronts that extend down and cover the exposed saws.

27 The saw teeth in traveling under the cotton roll take hold of some of the fiber and carry it between the ribs, leaving the seed in the roll box to be shed out as above explained. Each saw tooth that is loaded with lint cotton, amounting to from 0.000001418 lb. to 0.000001194 lb. per tooth from brush gins and

from 0.000001084 lb. to 0.000000934 lb. per tooth for air-blast or pneumatic gins, being 25 to 40 fibers per tooth based on 29,000,000 fibers per pound, must be unloaded or cleaned before it returns to the roll box again. The removal of the lint from the saw teeth is accomplished by means of a brush, Fig. 10, consisting of a series of $\frac{3}{4}$-in. by 1-in. sticks, usually 22 in number, in which bristles are inserted, usually projecting 1 in. from the stick. These sticks are mounted on a series of drum heads on a shaft $1\frac{15}{16}$ in. to $2\frac{3}{16}$ in. in diameter. The outside diameter of the brush cylinder is usually 15 in. Each brush has four wings 1 in. by 6 in. on each end head. In the operation by the brush gin the brush is set up to the saw cylinder so that the tips of the bristles overlap the saw teeth just $\frac{1}{16}$ in. and are then caused to rotate in the reverse direction to that of the saws, and at a peripheral speed of 6300 to 6900 ft. per min. or approximately six times as fast as the saw-teeth travel. When the bristles are new they have a wiping effect on the saw teeth, but after considerable use the tips wear off to such extent that they do not touch the saw teeth, and it is only the air blast generated by the brush that removes the lint. At the rear of the brush there is a board or sheet-metal cut-off, similar to that of a fan, and just above and between the saw cylinder and the brush there is a parting board that directs the air current on to the saw teeth, removing the lint and at the same time deflecting the motes out of the general current by centrifugal force with the assistance of the mote board, while the lint is blown into the lint flue below. The mote board is just below and between the saw cylinder and the brush, and is adjusted so that in the regulation thereof the motes just pass out while all of the lint cotton is caught in the lint flue. The usual air current or pressure produced on the saw teeth by the brush varies from 5 to 6.5 in. of water, and in its travel to the lint-flue main body it expands until the pressure in the flue is $\frac{3}{8}$ to $\frac{1}{2}$ in. of water.

28 The discovery that the air blast generated by the brushes had such an important part in removing the lint from the saw-teeth led to the discovery of the pneumatic gin. Figs. 11 and 12 show two different types of this gin. One (Fig. 11) discharges the motes below, while the other, Fig. 12, discharges them over the top of the air lip. In the early air attachments the discharge-nozzle opening was $\frac{7}{16}$ in. to $\frac{1}{2}$ in. wide and extended 1 in. beyond each end saw; after continued development, however, these openings were reduced to $\frac{1}{4}$ in. to $\frac{5}{16}$ in. The air blown from the lip

opening continues to hug the sheet that leads to the lint-flue opening, and this sheet is brought up to within $\frac{1}{32}$ in. to $\frac{3}{32}$ in. of touching the saw teeth. The lint opening is just wide enough to receive all the air discharged from the air nozzle, and in addition thereto a small amount of the outside air is injected by the dynamic action of the air current. There is thus a slight current from the outside leading into the lint flue, to insure that there will not be a loss of lint fibers on the outside, which would be the case if the

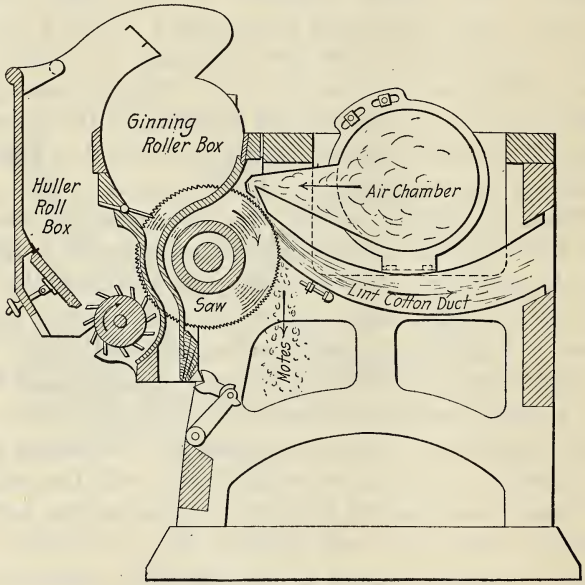

Fig. 11 Air-Blast or Pneumatic Gin

reverse action took place. The air pressure at the nozzle outlet is usually from 8 to 12 in. of water. The various pressures for other points in and around the air nozzle for the two types of gins considered are as shown in Figs. 13 and 14. The pressure in the lint flue at the rear of the gins varies from 0.7 in. to 1.0 in. in the rear of the gins to 0.5 in. to 0.65 in. in the lint-flue uptake to the condenser.

THE HULLER GIN

29 The operation of the huller gin is very similar to that of the plain gin, in that the saw cylinders and the methods of re-

moving the lint therefrom are the same. The roll-box construction is also the same except that there is an additional huller roll box in front of the ginning roll box, Fig. 5 and Fig. 9, for separating the seed cotton from the burrs, sticks, trash, etc., and the cotton is delivered into the huller roll box first, Fig. 5, and from there is carried up into the ginning roll box by the saws; hence the feed to the ginning roll box is from below instead of from the top, as is the case with the plain gin. The huller roll box, Fig. 5, consists of a

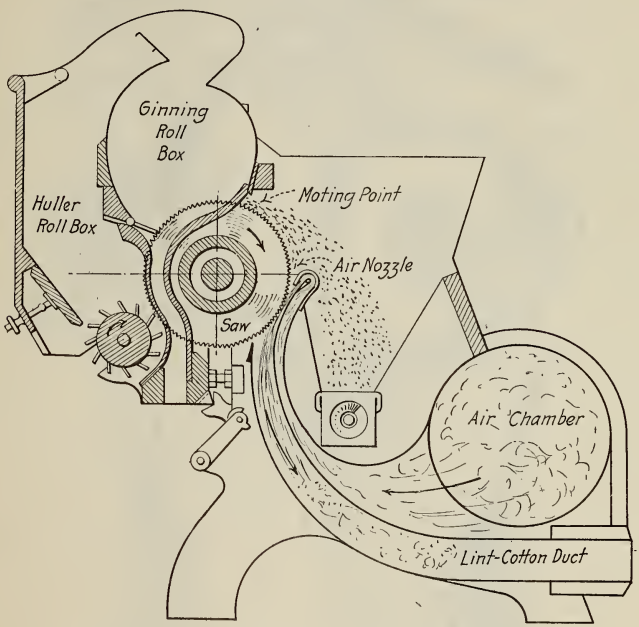

Fig. 12 Another Type of Pneumatic Gin

chamber in which the cotton is fed, and this chamber has a hinged removable front cover *a*, with an inclined sheet or board for regulating the refuse discharge, and a huller picker roller *b*, $2\frac{1}{2}$ in. to $4\frac{1}{4}$ in. in diameter, with 10 to 16 longitudinal rows of No. 8 spikes set at an angle of 30 deg. to radius of point where they enter roller. These spikes are alternated and in circular rows of 5 to 8 spikes between each pair of saws, and project out $\frac{3}{4}$ in. to 1 in. from the roller, which rotates at from 300 to 700 r.p.m. There is also a set of huller ribs, Fig. 7, to prevent the hulls and trash from being carried into the upper or ginning roll box, thus performing a double

ginning process. The huller ribs partly surround the huller picker roller, Fig. 9, and the outer or working surface of the rib is $\frac{1}{4}$ in. to $\frac{3}{8}$ in. from the tips of the saw teeth; just below the seed grate the rib curves out and is fixed to a rib rail or bar.

30 In the operation of the huller gin the cotton, Fig. 5, is fed

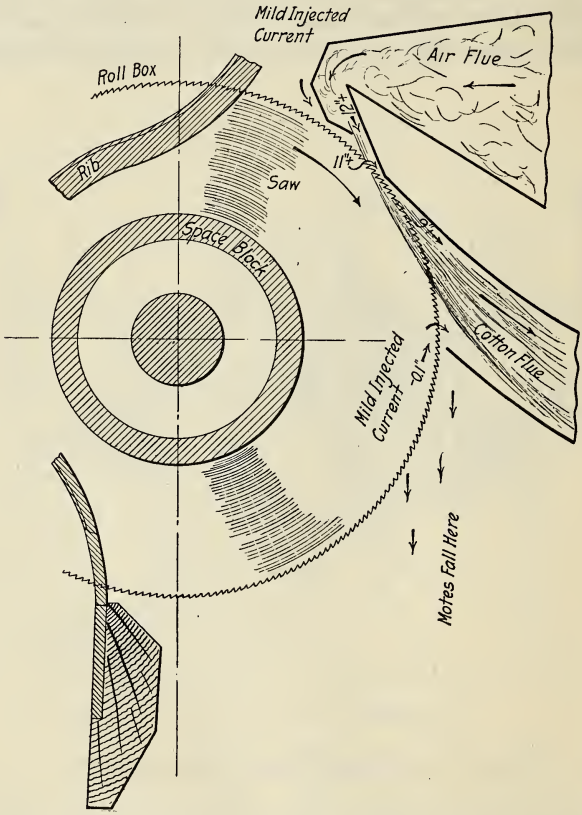

Fig. 13 Air Pressure at Various Points in Pneumatic Gin Shown in Fig. 11

into the huller roll box, the spiked picker roller rotating in the reverse direction to that of the saws, spikes inclining backward. The cotton is thus thrown on to the saw teeth and carried up into the ginning roll, leaving the trash, a part of which is shed out; but as the quantity increases, the box front is swung out by hand, discharging the foreign substances. When there is a large amount of trash the picker roller is pulled out from the ribs and the inclined

board moved back, thus permitting the trash to shed out in front of and behind the roller; but if there is only a small quantity, the huller roller should be set up close to it, to prevent the dropping of locks of cotton. The huller ribs are spaced sufficiently apart at

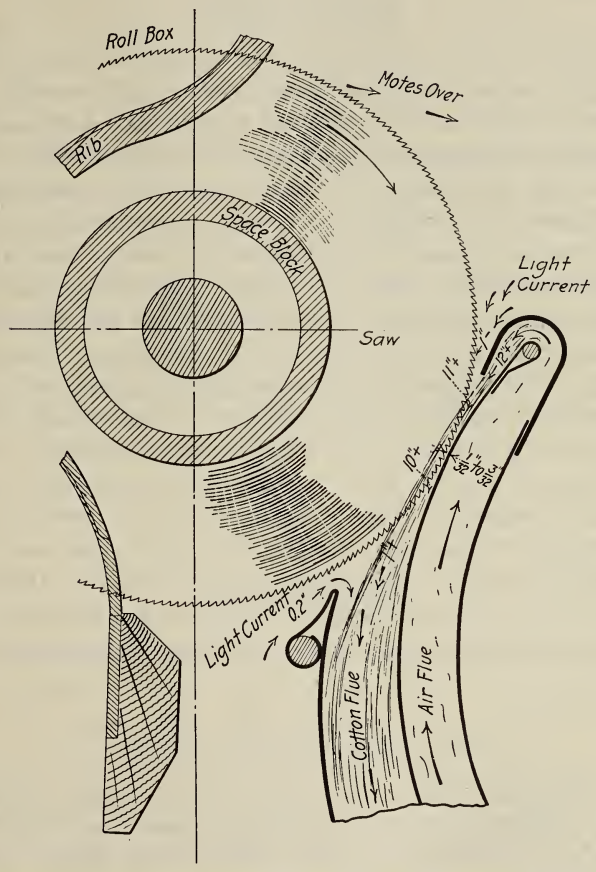

Fig. 14 Air Pressure at Various Points in Pneumatic Gin Shown in Fig. 12

the point where the teeth pass between to permit the seed cotton to pass through into the ginning roll box. The lint is removed from the saw teeth by means of a brush or current of air as has been described for plain gins, and the operation of the ginning roll box in shedding the seed is the same. The lint cotton is removed from the saw teeth and passes into the lint flue which has inlets from the

various gins set in battery. The size of the lint flue varies according to the number and size of gins, the general practice being to have about three square inches per saw.

THE CONDENSER

31 The lint flue delivers the cotton into the condenser which consists of a large horizontal drum covered with 8-mesh No. 21 wire screen. The drum is usually 54 in. long, the same as the cotton bale, and its diameter is usually 30, 40, 50, 60, or 72, in. respectively, for 100, 175, 250, 325 or 400 saws. The screen drum, Fig. 15, is mounted in a housing that spirally circumscribes it with air outlets at the ends and two doffing rollers at the discharge side, the lower one having flaps of thin $\frac{3}{32}$-in. light cloth-backed rubber. The rollers receive the lint cotton from the drum and deliver it on the lint slide and then into the receiving chamber of the press box. The portion of the screen drum exposed to the cotton air current is about 75 per cent, and there is 1 sq. ft. of screen provided for every 5 to 6 saws; but when the drum rotates it travels at a rate of speed of 50 to 80 ft. per min. for the smallest up to the largest-size condenser. This is equivalent to a screen exposure, when running, of from 2 sq. ft. down to 1 sq. ft. per saw per min. The peripheral speed of the doffing rollers should be the same as that of the screen drum. From the condenser the lint cotton is delivered by the doffing rollers to the lint slide, thence into the press box.

THE COTTON-GINNING PROCESS

32 The seed cotton is brought to the cotton-gin plant in quantities of 1600 lb. when clean-picked cotton and of 2000 lb. to 2200 lb. when gathered or grabbed cotton. When ginned there will be a finished bale weighing about 500 lb. and 1050 lb. of cotton seed; the remainder is refuse, as trash, hulls, dirt and other foreign substances. The seed cotton is usually hauled in wagons from which it is sucked off through a telescope by means of air-suction steel-plate centrifugal fans operating at a blast-wheel peripheral speed of 11,000 to 13,000 ft. per min., and producing a vacuum of 14 to 16 in. of water. In the course of its travel the cotton is carried through a cleaner where it is agitated on a screen to remove the foreign substances; thence it passes to the elevating system where

the air is separated from it, and is dropped into the gin feeders, where further cleaning is performed. The cotton is next delivered to the gin in such quantity as will not gorge the roll box, but will be sufficiently to insure a nice, moderately soft, pliable ginning roll. If the cotton is clean picked it can be delivered direct to the ginning roll in a plain gin; but if gathered or grabbed, it will contain hulls, etc., and should be delivered to a huller roll box.

33 The seeds fall from the roll box down into a seed conveyor from which they are delivered to the customer's seed box or into a seed storage house. The hulls, trash and foreign substances are conveyed outside of the gin house to the boiler room for fuel, — that is, that part that is combustible, — or hauled off to be used later as fertilizer. The motes are also conveyed out and in some instances used for low-grade fiber stock. The lint cotton passes from the gin to the lint flue and then to the condenser, where it is separated from the air and formed into a bat. It is then delivered into the press box for packing, wrapping and pressing, ready for the market, and after marketing is compressed and rewrapped before export.

DISCUSSION

A. W. Merkel. We have on file a copy of the drawing covering Eli Whitney's patent, and it shows very clearly the design as produced by him. This drawing bears date of March 14, 1794, which is presumably the date the patent was issued.

Whitney, like most inventors, never realized much in a financial way from his patent, because other parties began to manufacture the cotton gin, and it cost him very dear in time and money to carry his fight for justice through the courts. Finally becoming discouraged he left the South and returned to New England where he achieved success in other lines of manufacture.

All gins made for a number of years were of the plain gin type, as shown in Fig. 4.

The first attempt at making a gin to separate the hulls was in 1858. These first huller gins were of the single-rib type, and it was not until 1879 that the double-rib huller gin appeared. The first successful double-rib gin was brought out in 1879 by F. E.

Smith, a New Englander who had moved South and was at that time associated with the Daniel Pratt Gin Co., of Prattville, Ala.

The single- and double-rib huller gins accomplish the same results, that is they remove the hulls and trash from the seed cotton, but the double-rib gin has the advantage in that it will separate the hulls from the seed and discharge them into different receptacles, while the single-rib gin discharges them together.

The rib of the single-rib gin has a projection which extends towards the front, and which takes the place of the outside or hulling rib of the double-rib gin.

The pneumatic gin, described by the author, appeared in 1893, and was patented first by R. King of Mansfield, La.

It is interesting to note that although there have been numerous improvements in pneumatic or air-blast gins, as they are generally known, the original idea of Eli Whitney remains the best way of ginning cotton.

There is practically no difference in the rate of ginning, or in the sample produced by the air-blast and brush gins, on good cotton, but the latter has the advantage in the power required. For instance, in a four 70-saw outfit, which has four 70-saw gins, the four brushes will require 10 hp. to drive them, while with four 70-saw air-blast gins there will be required a 35-in. fan which will require 18 to 20 hp. to produce a pressure at the nozzle of 12 in. of water. This shows a saving of 8 to 10 hp. in favor of the brush-gin outfit.

The air-blast has the advantage of requiring fewer belts, and not having a brush to get out of balance, but this is offset by the fact that the brush will clean the saws better when ginning green or damp cotton in spite of the statement that the bristles soon become worn so that they will not touch the saws and that it then acts as an air-blast gin. If it does it produces a blast which hits the spot better than that produced by the fan.

Cotton gins are classed as "agricultural implements," which as a class do not require very close machine work, but the ribs, saws and space blocks which hold the saws apart must be accurately machined. The ribs must be carefully ground and polished so that they will allow the lint cotton to pass between them, and at the same time prevent the seed passing through. The polish is to prevent the lint being damaged by hanging to the ribs and to allow it a smooth, easy passage. The saws must be trained to run

in a vertical plane, because if they were to rub against the ribs they would damage the lint and probably set fire to it. The space blocks must be accurate to the thousandth of an inch, in order to bring the saws exactly in the center of the space between the ribs. All this work must be done by men who have had long experience at this line, and it takes several years to become proficient in the art.

In regard to the present method of punching saws making defective teeth, it is our opinion that punching and filing make a better, smoother tooth than milling. The writer has seen some milled teeth and has noticed that the milling cutter leaves a worse burr than does the punch, and since it would be necessary to file the burr in either case, it seems that punching is the better proposition, as it is very much faster than milling.

Gin saws are made in two thicknesses, 0.035 and 0.037 in., most of them being 0.037 in. thick. They are made of English and American steel.

Before the late war English steel was used almost exclusively, as the American steel was a very inferior product, but during the war as it was almost impossible to get shipments of English steel, the American firms began to improve their product, so that now, the American steel is just as good as the English.

In enumerating the kinds of power used to press cotton, the author did not mention hydraulic. This power is the most popular at present, and the general practice is to use an $8\frac{3}{8}$-in. diameter ram with 1800 to 2000 lb. per sq. in. pressure. This same power is also used to tramp the cotton and for this purpose a $3\frac{3}{4}$-in. ram with the same maximum pressure is used, but it is seldom that the maximum pressure is required for tramping.

To produce the pressure there is used a triplex hydraulic belt-driven pump, protected by a relief valve to prevent bursts, should the operator allow the pressure to run too high.

The horsepower required to gin cotton varies considerably. For instance, a four 70-saw outfit with 12-in saw gins, ginning at the rate of 500 lb. of lint cotton in 15 min., will require about 60 hp., while if the rate of ginning is increased to 500 lb. in 12 min., the power required will increase to 70 hp., an increase of $16\frac{2}{3}$ per cent.

The first successful elevator for elevating cotton from the wagon was produced in 1879 by the man who now heads one of the large gin manufacturing companies. The most efficient ele-

vating system of today is the one that makes use of a belt to distribute the seed cotton to the different feeders.

The combined output of all the cotton gin factories is estimated to be approximately 6000 gins per year. In normal times it is no trouble for the trade to absorb this number, because the ginning machinery is operated by negroes who give it such poor care that in a few years the gins must be repaired or discarded. The writer has known of gins that have given good service for 20 to 25 years, but that is the exception to the rule. The weak parts are essential parts of the gin, and are the saws and brush which will rarely last more than two seasons without being repaired.

There are made every year a number of hand gins with 10 or 20 saws. These are manufactured for use in experimental stations in this country, but the larger part of those produced are exported to South America. They are used in the districts that have not been opened up by railroads, and in the mountains where all products have to be brought out on the backs of mules, or other similar conveyance. There is also manufactured a small press that will press a 100-lb. bale, instead of the standard, 500 lb., and this is used by the export trade for the same reason.

There are several things needed to improve the common practice in ginning cotton.

First, some inexpensive way to take the dust and fine lint out of the gin house, which is especially bad on the operators when ginning dry, dusty cotton.

Another is some way to improve the sample of cotton that has been blighted by the boll weevil.

The boll weevil will often puncture only one lock of a boll of cotton. This lock then fails to mature and is picked along with the other locks of this boll. It is small and hard and when it gets into the gin the saws tear it apart, but the fibers instead of being nearly straight are very kinky. These fibers are, of course, mixed with the others and as they are curly and kinky, they will show up in the sample, thereby giving it not only a bad appearance but also lessening its value.

As previously stated, the brush type gin is better for ginning under all conditions, yet to overcome one of its disadvantages there is needed a better brush than is commonly used, and one that can be cheaply built.

After the cotton is baled in the ordinary press it is repressed

or compressed before it is shipped any distance. There should be a strong, cheaply constructed press; strong enough to get the required density in the bale in the first pressing operation, and cheap enough to be installed in every outfit.

W. E. Caldwell. The paper covers the subject of cotton-ginning machinery in an admirable manner, especially from the viewpoint of the designer. However, it is to be regretted that space was not available for a more thorough treatment of the subject for the benefit of those responsible for the operation of gins.

Cotton-ginning machinery has reached a high state of development within recent years, so far as performing the service intended is concerned, but the reliability for continuous operation is far from satisfactory. Ginning plants operate at the lowest annual load factor of any plants of equal magnitude in existence and during the cotton season every idle moment is expensive for the owner. Nothwithstanding this, however, it does not appear as though much effort has been directed towards rendering ginning systems more reliable in operation. The usual troubles encountered by cotton ginners are due to the too frequent use of set screws and clamped joints by the gin manufacturers instead of keys; overloaded belts occasioned by the use of small pulleys and low belt speeds; and lastly, fans and their piping arrangement. There does not seem to be a fan on the market today which is equal to the service and abuse encountered in the average cotton gin. Present-day gin fans are an improvement over those in use ten years ago, at which time the ordinary steel plate exhaust fan was employed and usually operated above a safe speed to obtain the desired pressure or suction. The improvements which have been effected since that time were in the substitution of forward curved vanes and an increase in the number of vanes. Despite these changes which have improved the characteristics of the fans it is necessary to operate them at a speed so high that their destruction occasionally results, or the life of the runner, bearings and shaft is materially reduced. The writer recently witnessed the destruction of two new fans, supplied by a reputable manufacturer, while in operation at 10 per cent below nominal safe speed. This may be accounted for by the fact that manufacturers' shop speed tests are conducted with closed suction or discharge, hence there is but little pressure difference on the vanes to assist in starting the initial buckling, which

always precedes the ultimate destruction of the runner. The pressure difference while relatively slight is of some moment, especially with wide runners, since its effect is in the same direction as that of the centrifugal force.

With the larger plants employing batteries of five or six 80-saw stands it is customary to supply a size 45 or 50 fan for the blast air, operating at a peripheral speed around 13,000 ft. per min. to give the desired results. As the bursting speed of the usual runner for a fan of this size is in the neighborhood of 14,000 ft. per min. the writer feels that the margin of safety is too low for continuous operation and that it is possible to improve the runners by the use of stiffer and better construction to increase the bursting speed to beyond 17,000 ft. per min.

It is probable that in many instances equally good results may be secured by running the fans at a lower speed than that recommended by cotton-gin manufacturers, although operators of gins do not feel inclined to order the change until trouble actually occurs. It would seem to the writer that air-blast gins have been developed to accommodate the fans now on the market, whereas a fan should be designed better to meet the requirement of ginning systems. Perhaps it would be advantageous to employ a fan somewhat on the order of the cupola blower, which has a better pressure-volume characteristic than the usual volume fan, as well as greater efficiency, although the efficiency is relatively unimportant considering the low annual load factor of ginning plants. In any event it seems extravagant to advocate the use of a volume fan for this work, having an outlet opening over four times as great as the total blast area of the gins. It would be interesting to know the volume of air delivered per saw under normal operation, for a given air pressure in the main duct at the gin stand. The duct pressure is more easily measured than the nozzle pressure and the relation of this pressure to the air flow would provide useful information for determining the proper size as well as proper speed for the fan. The figures quoted by the manufacturers cover a wide range, but the average quoted is in the neighborhood of 20 cu. ft. per saw with a 12-in. nozzle pressure.

The author quotes the tooth travel of pneumatic gins at from 1400 to 1900 ft. per min. What determines this upper limit of saw speed? If the limit is imposed by the blast air velocity, could not this be increased by reducing the area of the blast opening

and employing a higher pressure, thus increasing the capacity of the gin?

The author describes the pneumatic elevator but fails to mention the belt distributor which is quite popular in many sections of the cotton region, and there are a few instances where it has replaced the pneumatic elevator after one or two seasons' operation of the latter. From the writer's point of view the pneumatic elevator is superior to the belt distributor, considering the power saving, greater simplicity, ease of operation and saving of time between bales. Has the belt distributor any real advantages over the pneumatic type or is the selection merely a matter of personal preference?

It would be interesting to know the power consumption of different-sized outfits and whether or not the air-blast systems are as economical in this respect as the brush type. The power consumption is an important item when contemplating the installation of an oil-engine drive, or other prime mover having limited overload capacity. It is appreciated that the power consumption of the fans will vary with different installations, but if the author is able to itemize the consumption of the different pieces of apparatus under average conditions it will place valuable data before the members of the Society.

No. 1840

MATERIAL-HANDLING EQUIPMENT AS USED IN THE IRON AND STEEL INDUSTRY

By F. L. Leach, New York, N. Y.
Associate-Member of the Society

This paper describes the handling machinery and apparatus used in the manufacture of steel. From the time that the ore leaves the mines until the steel goes through the last process at the mill it is moved about exclusively by different types of heavy machinery designed especially for the purpose. The author expresses the hope that through his description, weak points in present-day practice will show themselves and means of improvement be suggested.

THE iron and steel industry probably requires a greater diversity of material-handling equipment than any other type of manufacturing, because of the enormous bulk and weight to be handled. From the time the material leaves the mines until it is turned out of the mills as finished product it is constantly being moved by heavy conveying machinery of all descriptions. It is the intention of the author to describe this equipment with the idea of bringing out weak points in present-day practice, and possibly open the way to strengthen the weakest links in conveying methods used.

2 As the principal object of this paper is to describe the machinery used about the industrial plant, just a brief description will be given of the handling methods before the material reaches the plant.

3 At the present time ore handling on the Great Lakes is carried on by a fleet of vessels unequalled anywhere in the world. Some of these vessels are capable of handling a cargo of 15,000 tons, and are over 600 ft. long. During the year 1916, 66,658,466 tons of ore were carried by the vessels from the upper Great Lakes to

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southeastern ports. The steamer *Homer D. Williams* carried 462,490 tons during the season of 1920.

4 Vessels equipped for self-unloading have been developed for handling stone and ore. The hold in such cases is divided into bins, which deliver the material by gravity to a belt conveyor running lengthwise of the vessel; this conveyor delivers the material to an elevator which hoists it to another belt conveyor suspended in a horizontal position from the masts of the vessel and capable of being moved radially and vertically like the boom of a derrick. This belt conveyor delivers to the storage pile on the dock, and it is

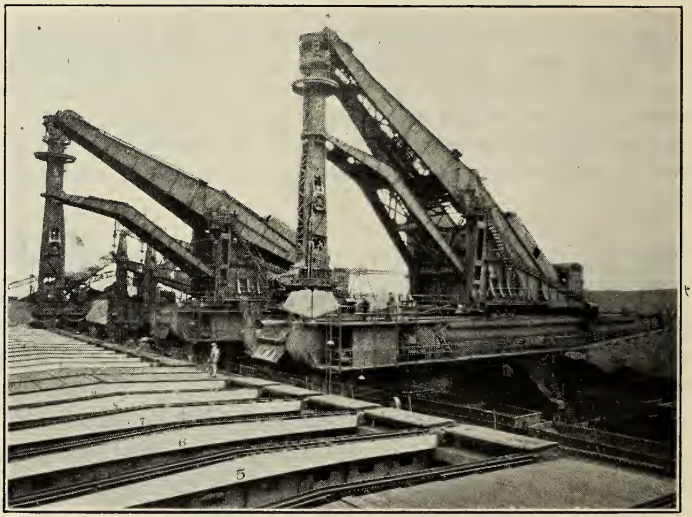

Fig. 1 Hulett Ore Unloader at Ashtabula Harbor

possible to distribute the material onto the dock at any point or height desired within the scope of the boom conveyor.

5 Ore-unloading equipment has probably been developed to the highest point of efficiency at the ports on the Great Lakes. An example of this is Ashtabula Harbor on Lake Erie, where there are eight Hulett ore unloaders having a capacity of 15 tons each. They have a record of unloading seven boats of a total capacity of 70,000 tons in 22 hours actual time. Four 17-ton capacity machines of this type have unloaded a cargo of 13,000 tons in 3 hours and 25 minutes. Fig. 1 shows a view of these machines in operation.

6 These unloading machines remove the ore from the boat and dump it directly into railroad cars for shipment to inland plants, or they place it in a bin located at the rear of the machine where it may be picked up by a traveling ore bridge and placed in storage piles.

7 Ore shipped to inland plants in bottom-dumping cars may be unloaded directly into bins by gravity. However, the modern method is to use a traveling rotary car dumper as shown in Fig. 2 which can be spotted at any point in the length of the ore yard. The cars of ore are placed one at a time on the

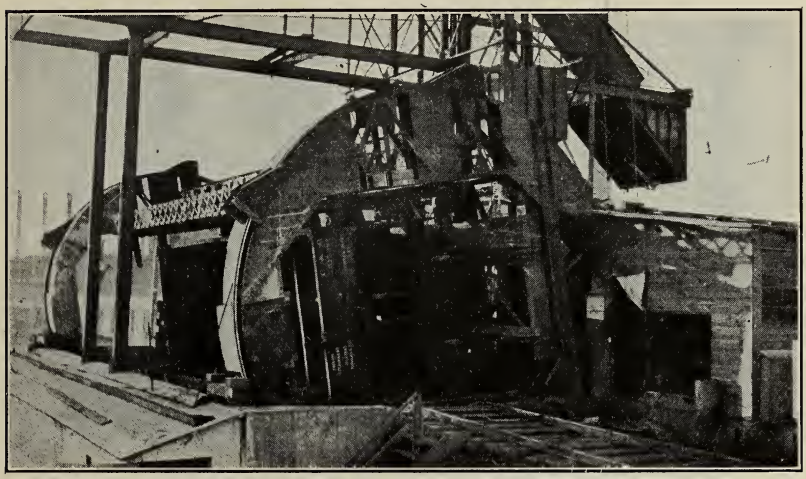

Fig. 2 View of Rotary Car Dumper

dumper, turned upside down and the ore dumped into a concrete pocket that extends the full length of the ore yard. An ore bridge distributes the ore to the storage pile in the yard. Rotary car dumpers have been developed so that three men can unload about 20 cars per hour. Cars of 240,000 lb. capacity have been designed with a total dead weight of 78,800 lb., giving a ratio of revenue load to total weight of 75.4 per cent.

8 The traveling ore bridge is another valuable link in material-handling equipment and is used in many different ways in handling the ore at unloading docks. One of the largest bridges of this type ever built is 612 ft. long overall, and is used at the Western Pennsylvania Dock Company at Cleveland, Ohio. This

bridge handles a 15-ton bucket and covers an ore storage of over a million tons.

9 These bridges may also be used to place the ore and limestone in bins, preparatory to sending them to the blast furnace. Several different types of bins are used, although the principle is the same in each. The ore is placed in the bin either by the ore bridge, or by gondola-type hopper cars and it passes down through the bin by gravity as required, through a shut-off gate into a lorry car. From the bins an electrically driven

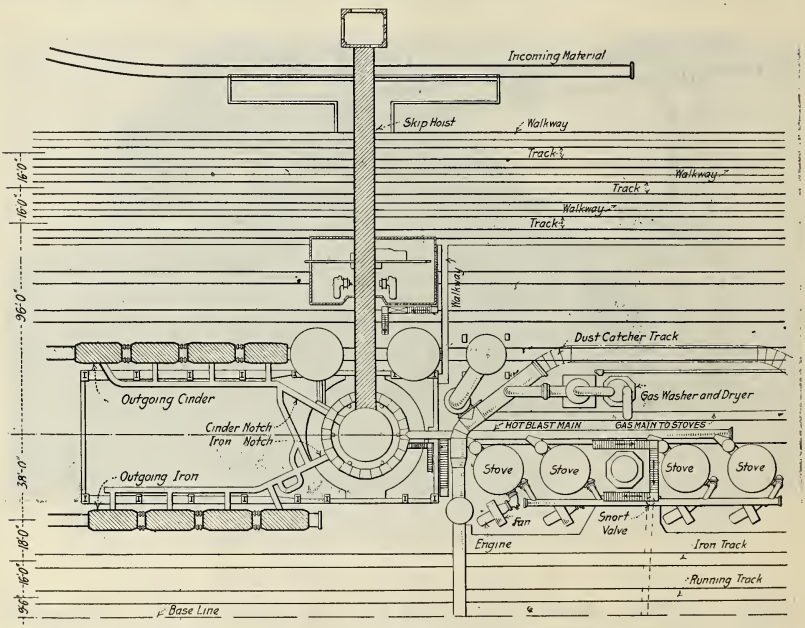

Fig. 3 General Plan of Blast Furnace and Stoves

weighing lorry car is used to convey ore, limestone and coke to the blast-furnace skip hoist.

10 Fig. 3 shows how material is conveyed from the stock yard and bins to the top of the blast furnace, dumped into the bell hopper by the skip car, and finally deposited inside by the well-known double bell system. Two of the products of the blast furnace are pig iron and slag, which are drawn off at the base in a molten state. Both iron and slag were formerly allowed to run

from the furnace into sand molds and handled after cooling. Most plants today, however, run the molten iron and slag directly into ladles supported on special cars. In this way the iron is transported either to a pig-casting machine where it is cast into pigs, or to a mixer at the open hearth plant, where it is kept in the molten state, or further refined before it is placed in the bessemer converter, for the duplex process. The molten pig iron may also be conveyed direct to the open-hearth furnace for the straight open-hearth process.

11 The slag is disposed of in various ways, since it has become of considerable commercial value in the last few years. After

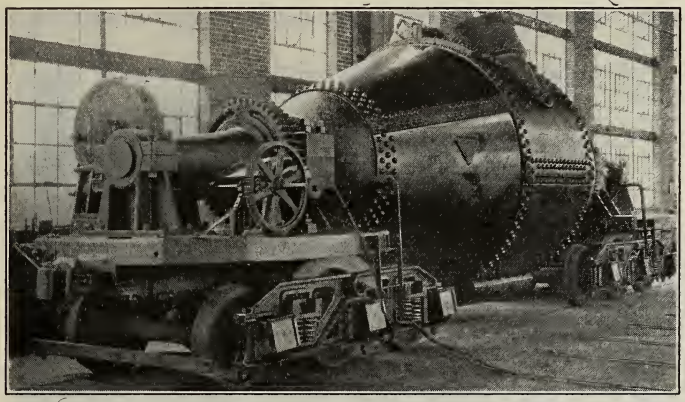

Fig. 4 Pugh Mixer-Type Hot-Metal Car

having been allowed to cool it is crushed to various sizes, and then used in the place of crushed stone.

12 A new design of car known as the Pugh mixer-type hot-metal car, shown in Fig. 4, has recently been developed for handling hot metal from the blast furnace. This car, built in capacities ranging from 70 to 150 tons, is claimed to have many advantages over the ordinary type, chief among these being the heat-saving feature which allows the hot metal to stand in the car for a period of 36 hours without "freezing." It is also a labor saver because, due to its large capacity, it requires less runners to distribute the metal from the furnace to the cars, thereby eliminating considerable scrap.

13 Fig. 5 shows a view of a typical slag car for conveying the molten slag away from the blast furnace. It is dumped pneumatically.

THE DUPLEX SYSTEM FOR MAKING STEEL

14 Fig. 6 shows the plan of a modern duplexing system for manufacturing steel. The machinery in this department of steel making must stand the most severe treatment of all, as the steel is handled at temperatures ranging as high as 2900 deg. fahr. It can therefore readily be understood how important it is to have such machinery and equipment designed as perfectly and as safely as possible.

15 The hot metal from the blast furnace is brought to the mixers by ladle cars. On Fig. 6 this car is shown in position before

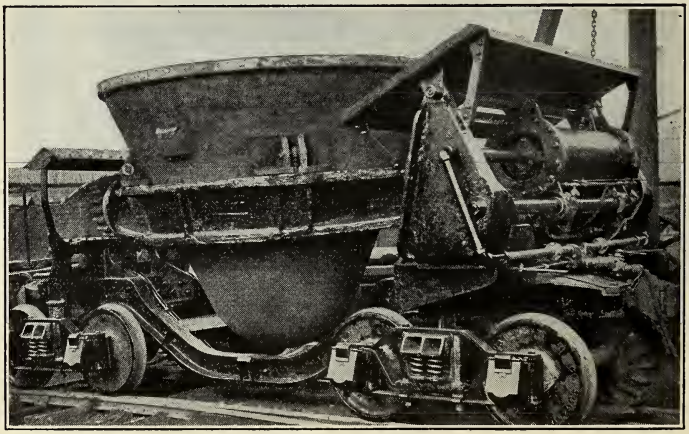

Fig. 5 Slag Car and Ladle

a 1300-ton-capacity mixer into which the content is poured by lifting the ladle free from the car with the aid of a 100-ton ladle crane.

16 The hot metal is removed from the mixer as required, by tipping it so that metal will run out of a spout opposite the receiving side. Another ladle supported on a ladle car is used to transfer the hot metal from the mixer to the bessemer converter. This ladle is tilted by a jib crane and the contents poured into the top of the converter already tilted to a receiving position.

17 The next step for the hot metal in the process of refine-

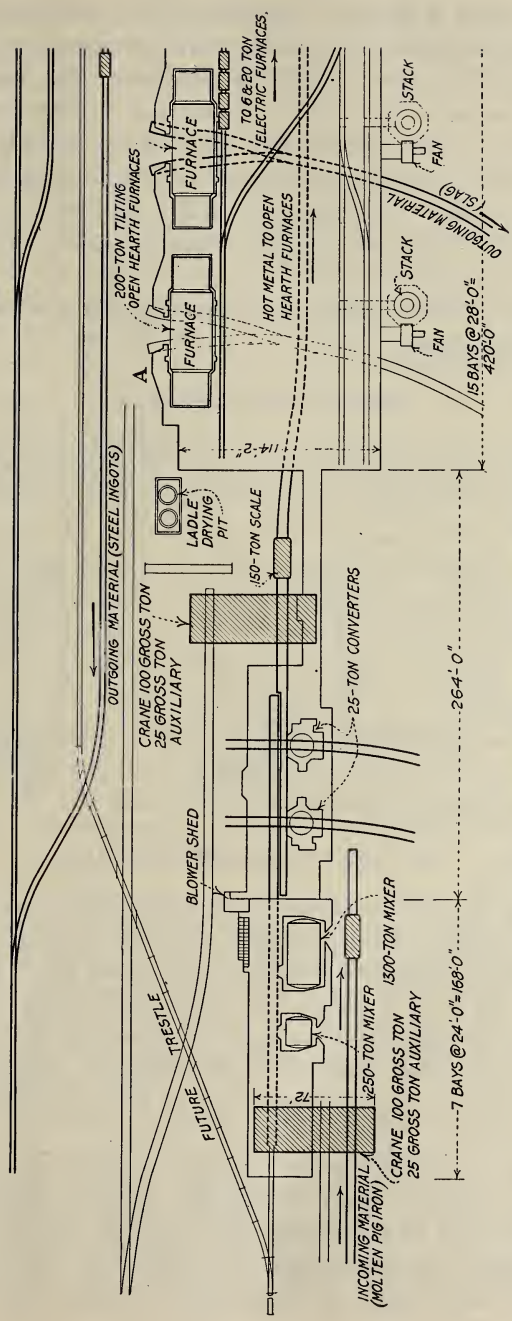

Fig. 6 General Plan of Duplex Plant

ment is to convey it from the converter to the open-hearth furnace. This is done very much as was the previous operation except that the hot metal is poured from the converter into another ladle mounted on a ladle car and is transferred by steam or electric locomotive to a point before the open hearth furnace. Here, it is hoisted free of the car and poured into the furnace by means of an overhead crane. The track over which this car travels is indicated on Fig. 6 by the legend "Hot Metal to Open-Hearth Furnaces."

18 Fig. 7 shows a 65-ton ladle car that may be used for this transferring operation.

OPEN-HEARTH FURNACES

19 Scrap, ore and limestone are conveyed to the open-hearth furnace in cast-steel boxes called charging boxes, three or four of

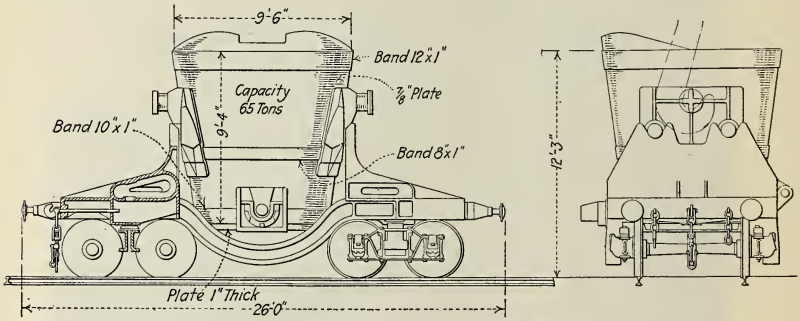

Fig. 7 65-Ton Transfer Ladle Car

which are moved on a specially built car. The scrap is placed in the boxes by means of the overhead crane and a magnet. The ore and limestone are handled either by the crane and a grab bucket in cases where it is necessary to move them from storage piles to the boxes, or they are placed in the boxes direct from bins by means of chutes, when the plant is so provided. The cars are then hauled in trains to the charging floor of the open hearth furnace building.

20 It occurs to the writer that the ore and limestone used in open-hearth practice could be handled more economically if it were either stored in bins under the approach trestle to the open hearth building and discharged by chutes into the charging box cars, or if it were stored in bins under the open-hearth floor and

removed by means of a grab bucket, or possibly by the application of mechanically operated proportioning devices used in conjunction with compartment bins discharging same to the charging floor of the open hearth.

21 In the methods now in vogue, the train of loaded charging boxes is transferred from the storage yard to the open-hearth charging floor in front of the furnaces by means of an electric or steam locomotive. The material in the charging boxes is then dumped into the furnace by an electric charging machine. This machine lifts the charging box into the furnace where it dumps its contents by turning the box through a complete revolution. The operator of one of these machines sometimes takes a loaded box and uses it as a means of leveling the charge within in the fire bed of his boiler.

22 The products of the open hearth are steel and slag. The steel is drawn off from the furnace and run into a ladle of a size consistent with the capacity of the furnace. In the Talbot system only half of the charge is drawn from the furnace at one time. After the ladle is filled it is transferred by an overhead crane to a position over the first mold of a train of ingot molds.

23 The slag which flows off at the time that the steel is tapped, is run into a slag ladle mounted on a special car by which the ladle is carried to a suitable point for dumping. The position of this car is indicated on Fig. 6 by reference (A). It is placed there to receive the slag during the tapping of the furnace and is hauled away afterwards on the track indicated by the note.

24 The molds in which the steel is cast are mounted on cars usally three or four molds to the car. These cars are heavily constructed as they must withstand having molten metal splash on all parts of them or the dropping of an ingot from a height of several feet when the mold is being stripped from it.

25 The direction of travel of these ingot cars is indicated by arrow heads and notes, as they travel from the open-hearth pit to the stripping yard.

26 The stripping process simply means removing the molds from the cast ingots after the metal has hardened sufficiently. Fig. 8 shows an ingot stripper, and indicates how the hooks of the stripper engage the ears of the mold lifting it clear of the ingot car and leaving the ingot on the car. When the mold has been used for a long time it may become slightly pitted, causing the ingot to stick

when it is being stripped from the mold. Under these circumstances it is necessary to drive the ram down into the opening at the top of the mold to force out the ingot.

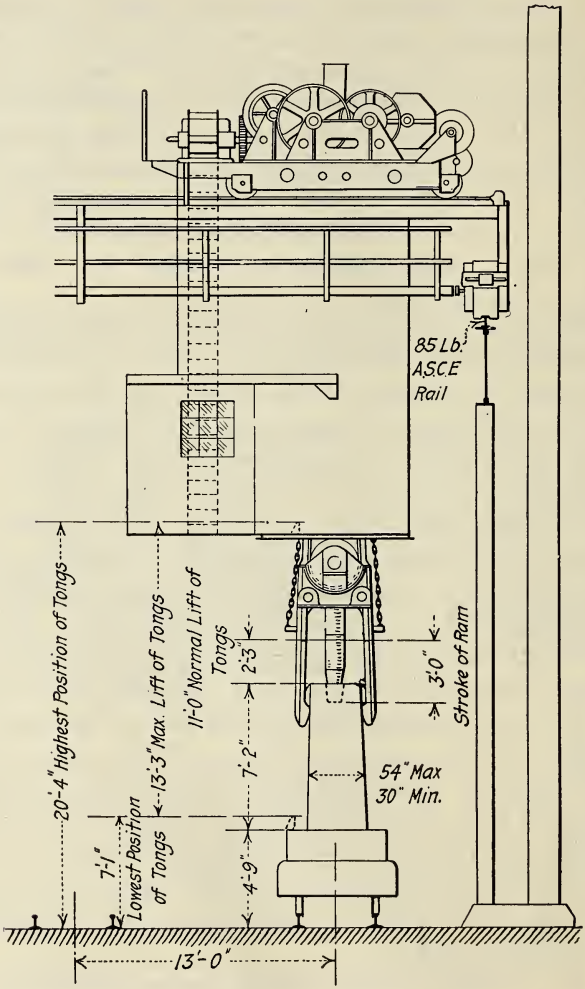

Fig. 8 200-Ton Ingot Stripper

27 The train of stripped ingots is now moved from the stripper yard to the soaking-pit furnaces, or to a suitable storage yard.

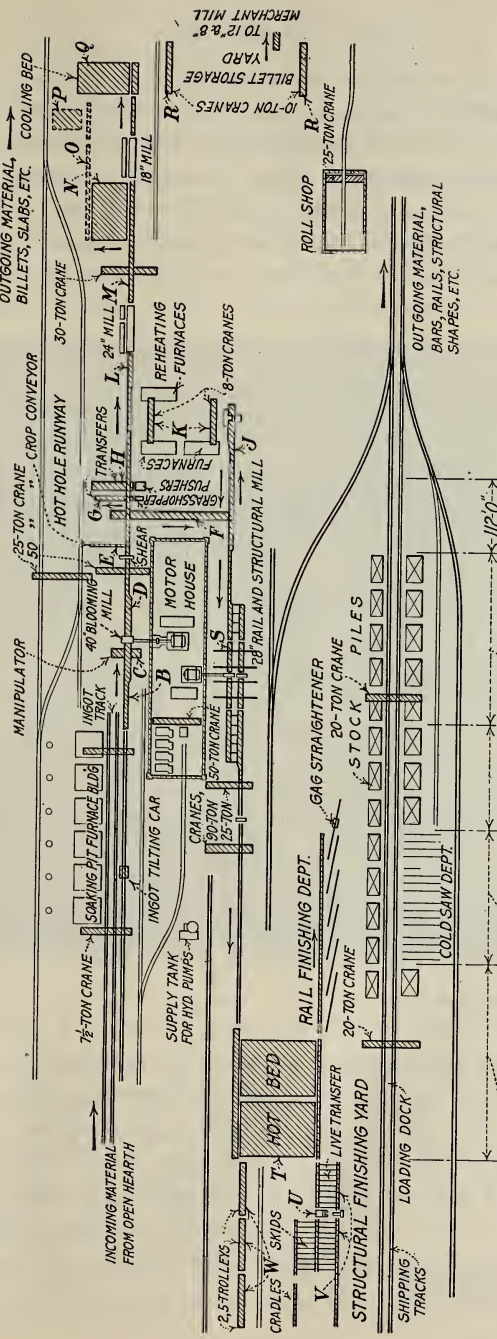

Fig. 9 General Plan of 40-in. Blooming Mill and Sheet Bar and Billet Mill

MODERN ROLLING MILL

28 Fig. 9 shows a plan of a modern rolling mill, in which nearly all the different types of material-handling machinery that have been developed to the present time for this industry are used.

29 This plant includes a 40-in. two-high reversing blooming mill, a 24-in. and 18-in. sheet bar and billet mill, and a 28-in. rail and structuarl mill. Provision has been made for serving a 12-in. and 8-in. merchant mill with billets from the sheet bar and billet mill. All mills are electrically driven. The different types of conveying machines are indicated by letters, in the order in which the material travels.

30 The blooming mill is served by six 4-hole soaking pits, in which the ingots are placed by an overhead electric tong crane. This crane handles the ingot on a principle similar to that of a pair of ice tongs. The operator's cage rides with and above the tongs, enabling him to place the ingots in the proper position in the pits.

31 The next step in moving the material is to serve the blooming mill with hot ingots from soaking pits. The soaking-pit crane delivers the ingot to a tilting buggy, places the ingot on roller table (B), and it is directed properly for entering the rolls of the blooming mill by manipulator (C).

32 Fig. 10 shows a general view of soaking-pit furnaces, and the end of a roller table, to which an ingot is being delivered by the traveling tilting buggy. This buggy travels back and forth in front of the soaking pits, operated either by a remote control switch which places the buggy at any desired position on the track, or motivated by an electric trailer — man-operated. In case of breakdown of the ingot tilting buggy the cranes can place the ingot directly on the table.

33 As previously stated, the manipulator is a machine used to place the ingot in the proper position for entering the rolls, and to turn or shift it for the next pass after passing back and forth through them. The modern manipulator is hydraulically operated. Its pushing arm has both a vertical and horizontal movement and is provided with a movable side guard on each side with tilting heads on one side for turning the section being rolled. This enables the operator to place the material at any desired position before the rolls.

34 Many mills are equipped with manipulators on both the entering and delivery sides of the mill, but when they are placed on delivery side the tilting heads are omitted.

35 After ingots have been reduced to blooms, slabs, or blanks by the blooming mill they are carried on to the shear by another electrically driven roller table indicated on Fig. 9 as table (D). The shear cuts the blooms to required lengths removing the fan-tail end or " crop end," as it is termed in the mill. This crop end is dropped down a chute at the shear to conveyor (E)

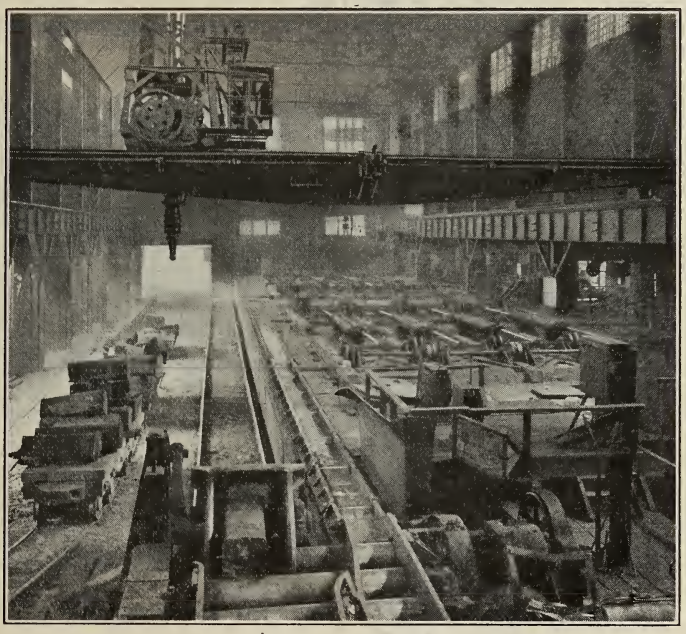

Fig. 10 General View of Soaking Pits, Ingot Tilter and Tong Crane

which conveys it to a bloom-butt car located on the track next to the mill building. This conveyor is usually very heavily constructed and due to the irregular shapes handled, it is built so that the billets are pushed along on a smooth surface of plates by a traveling scraper conveyor which will handle any size or shape liable to be produced by the mill.

36 In some plants open-hearth charging boxes are placed on charging car and the crop ends are loaded directly into the

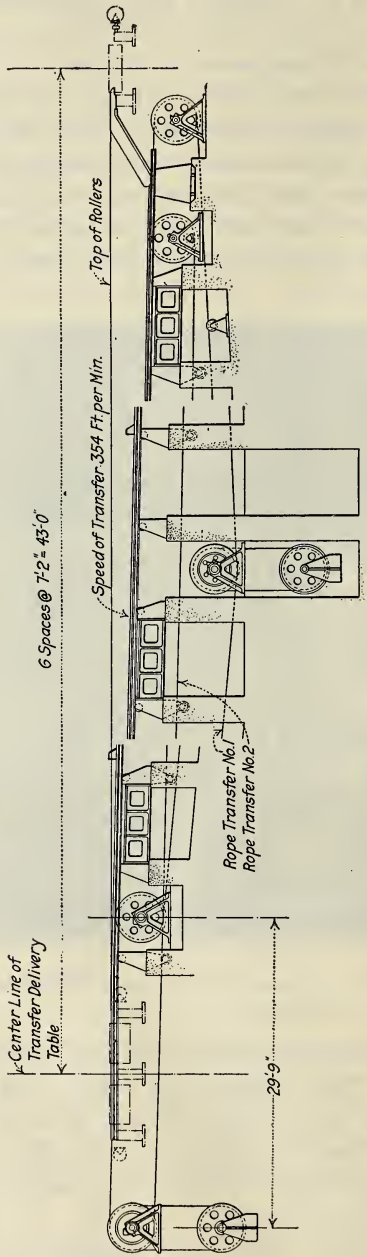

FIG. 11 145-FT. BLOOM TRANSFER

boxes by the crop-end conveyor. This is desirable where the distance between the blooming mill and the open hearth is not far. If the distance is too far, too much money is tied up in charging boxes, due to the quantity necessary to provide for delays in transportation, etc.

37 The mill layout shown on Fig. 9 provides for distribution of blooms and blanks to the rail and structural mill by means of conveyor (F) and slabs and blooms to the hot-hole or shipping yard by means of conveyors (G) and (H). It also provides for distribution of blooms and blanks to reheating furnaces or to rail mill by means of conveyor (F), table (J) and charging cranes (K)

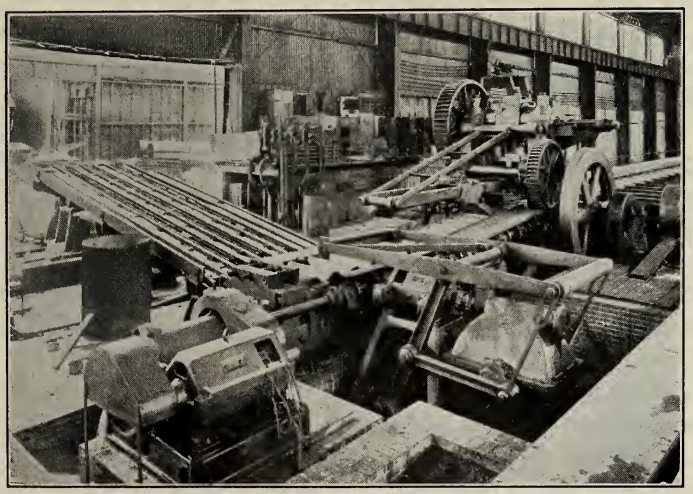

Fig. 12 Billet Conveyor and Grasshopper Pusher

when reheating for rail mill, or for blooms and slabs direct to the sheet bar and billet mill by table (L). Conveyor (F) shown in Fig. 11 is constructed with a surface of heavy tee rails on which the billets are skidded along by rapidly traveling dogs. The dogs run on tracks between the rails and are attached to wire ropes running around an idler pulley at one end of the bed and a driving drum at the other end. This conveyor must handle the blooms rapidly as they are sometimes passed from the blooming mill direct to the rail and structural mill without being reheated.

38 Fig. 12 shows another type of these conveyors and also the grasshopper type of pusher, a machine used for removing material from a roller table to a conveyor placed at right angles to it.

SPECIAL FEATURES OF GRASSHOPPER TYPE OF PUSHER

39 The feature of this machine is the peculiar path which the ram describes in passing through one cycle of operation. The levers are arranged to cause the ram to travel in practically a horizontal straight line in the forward stroke, while on the return stroke it describes an arc in an upward direction, clearing material that may be traveling out from the shear.

40 When it is desired to deliver material to the hot hole it is removed from the shear table by means of the grasshopper pusher, to either conveyor (G), Fig. 9, in the case of slabs, or to conveyor (H) in case of blooms or blanks.

41 When slabs are delivered to conveyor (G) it transfers them to a slab piler which arranges them in piles of four or five. They are then removed by the overhead crane to be stacked in the hot-hole yard and allowed to cool.

42 The same procedure takes place when blooms are delivered to conveyor (H), except that the blooms are delivered to a cradle from which they are removed to the hot-hole yard for cooling and storage.

43 The roller table indicated by reference (L) on Fig. 9 is the ordinary type of electric-driven table.

44 Many types of machines have been developed to replace the old-style hand methods of replacing heavy billets, bars, etc. in the furnace for reheating, preparatory to rerolling. Reference (K) on Fig. 9 indicates the location of two of these machines which travel on girders above the furnaces. They are designed so that the tongs may be placed in any position necessary to serve the furnaces on either side of the charging floor.

45 In the plant shown by Fig. 9 the charging crane (K), can pick up the material from the skids at the end of table (J) and place them in the furnace; store material at the end of the runway for future use; remove hot material from furnaces, and serve either the sheet bar and billet mill or rail and structural mill. This arrangement allows many methods of operation between the different mills.

SHEET BAR AND BILLET MILL

46 When it is desired to reduce the blooms to billets or flat bars they are conveyed direct from the blooming mill shear to the 24-in. mill by table (L) and rolled on the 24-in. mill without reheating. Some few sections, however, require reheating and the blooms for these are taken from table (L) by charging crane (K), placed in reheating furnaces, then returned to table (L), and to the 24-in. mill. After passing through the 24-in. mill they are delivered to the 18-in. mill by table (M) where the cross-sectional area of the bar is still further reduced. If it is not necessary to pass the bars through the 18-in. mill they are delivered to the stock yard or cars by means of a cooling-bed transfer conveyor (N), roller table (O) and transfer (P).

47 If it is necessary to pass the bars through the 18-in. mill they are conveyed to same by table (M). This table is designed with the axis of the rollers set at an angle of about 102 deg. with the center line of the table instead of 90 deg. (Fig. 15) for the purpose of throwing the bar off the table onto transfer conveyor (N). If it is desired to deliver the material to the 18-in. mill instead of placing it on transfer table (N) it is guided into a groove which is on the circumference of each roller, these grooves being in line on the table. Naturally this causes the material to follow a straight course to the 18-in. mill instead of being thrown off the table by the skewed arrangement of the rollers. After passing through the 18-in. mill the bars are cut to size by a rapidly operating shear, the knives of which move forward with the bar cutting it as it rolls along. This type of shear is called a flying shear. The bars are then conveyed to cooling bed (Q), where they are allowed to cool, and transferred by the cooling bed conveyor to a bar piler at the end of the conveyor. The bundles of steel bars or billets are then transferred to the billet storage yard, or to the merchant mill for further reduction to small shapes, or directly into cars for shipment by overhead crane (R).

28-INCH RAIL AND STRUCTURAL MILL

48 As previously mentioned it is possible to deliver blooms from the blooming mill direct to the rail and structural mill by means of conveyor (F), thereby continuing the reduction of the

bloom on the rail and structural mill without reheating. If it is desired to reheat the bloom, it is carried by means of conveyor (F) and roller table (J) to the end of table (J), where it is pushed onto the skid table, picked up by a charging machine (K), and charged into the reheating furnace. When properly heated it is returned to roller table (J) by charging machine (K).

49 Table (J) is designed with a double set of rollers so that blooms delivered to it may pass in either direction on both sides at once. The advantage of this can easily be understood; for example conveyor (F) may be delivering material to the furnace on one side of the table and at the same time material may be transferred from the furnace to the rail and structural mill on the other side. This arrangement is shown on Fig 11 by a light outline of the cross-conveyor at the delivery end of the conveyor.

50 If the bloom is to be rolled on the 28-in. mill it is carried to the mill by table (J) and a traveling tilting table (S). Two of these traveling tables are located on either side of the 28 in. mill and serve the purpose of catching the bloom as it travels back and forth through the mill. When the cross-sectional area of the bloom has been reduced to as small as is desired on the first stand, it is pushed over to the second stand by means of rope-propelled dogs which travel back and forth between the skid rails of a stationary rack transfer. The traveling tilting table (S), moves the material from pass to pass in either the top or bottom pass, as required. In event of either table breaking down, provision is made for removing the table beyond the face of the rolls where repairs can be made, and one table is used for both stands, which of course lessens production. When the roughing out is completed on the second stand of the 28-in. mill, the material is passed to the finishing stand of the 28-in. mill. Here it is given one finishing pass of either a rail or structural shape and passed on to the saws by another power-driven roller table.

51 If the finished material is a rail it is generally cut into three pieces of standard length simultaneously by four drop hot saws, the two end saws being used to remove the crop end of the rail. If a structural shape, it is cut by the sliding saw indicated to lengths desired. The material is then delivered to the cooling bed (T), on which it is drawn across by dogs attached to wire ropes or some other of the many methods of propulsion, that have been designed for this purpose.

52 Rails are delivered in one direction to a rail-finishing department and structural shapes in the opposite direction to a department for finishing I-beams, channels, angles, etc.

53 When finishing structural shapes they must be straightened and the ends cut true. Fig. 9 shows a modern arrangement for transferring shapes from the cooling bed to the straightener (U) and thence to the shears, by means of transfers (V). When shapes have passed through this last stage of finishing they are removed from the shear cradle by means of overhead cranes (W) and placed in piles for storage or directly in railroad cars for shipment.

54 Rails pass through a similar process of straightening and end finishing and are also drilled. They are conveyed from the cooling bed on roller tables, and transferred to the different finishing machines on idler rollers supported on individual stands.

55 The writer is of the opinion that the methods of handling rails in the finishing department can be materially improved and more of the manual labor eliminated by different plant arrangements than are used at the present time. Not enough power-driven machinery is used in this work. This problem could be studied with beneficial results.

56 The foregoing description gives a general idea of the type of material-handling machinery used in a steel mill, but of course does not cover all the variations that would naturally be required in manufacturing different forms of steel. For example, there are several types of mills which have not been mentioned, all of which require material handling equipment suited to the type of material handled.

57 The mill for manufacturing steel plates requires lighter but broader tables, and toward the finishing end chain conveyors are used to handle the plates. At the shears an arrangement of castors set up on spindles is used, allowing the plate to be placed at any desired relation with the shear. This arrangement is known as a castor bed.

58 The skelp and rod mill requires light tables and conveyors which will handle the rolled material rapidly, as the mills used for this work are small and run at a high rate of speed. They also require long cooling beds, that will handle the material in such a manner that it is straightened while cooling. The Edwards escapement bed is a good example of this type, and is

shown by Fig. 13. The rods are delivered to the top of the bed by a conveyor having a set of cone-shaped rolls. These throw the rod to the small end of the roll which retards the speed of the rod and allows a set of fingers to pick it up and place it on the first of a series of spurs on an inclined bed. As each rod passes out from the mill it is picked up and placed on this inclined bed, and as each is dropped on the bed the preceding rods are dropped down one spur at a time until they reach the bottom.

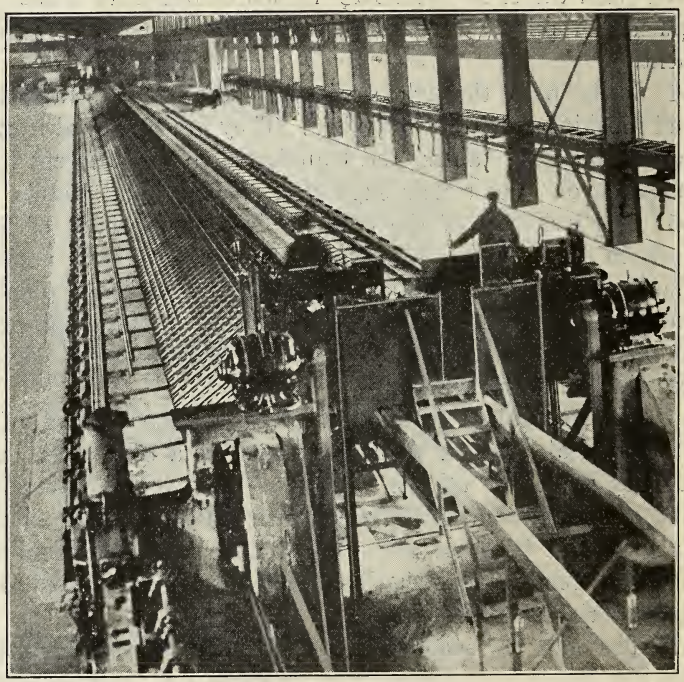

Fig. 13 Edwards Double-Escapement-Type Cooling Bed

This intermittent dropping from one spur to the other straightens the rapidly cooling rod so that when it reaches the bottom it is straight enough for ordinary purposes. The rod is then pushed out onto a roller table at the bottom of the cooling bed, and conveyed to the shear where it is reduced to the desired length. In some cases where the rods are small in diameter, for example $\frac{1}{4}$ in. and $\frac{5}{16}$ in., they are passed from the mill direct to coiling reels, where they are coiled into bundles and placed on a slow moving coil conveyor. This allows the bundles to cool before being placed in storage piles or on cars for shipment.

SHEET AND TUBE MILLS

59 Although material-handling equipment in sheet mills has improved considerably in the last few years, it is the opinion of the writer that there is greater chance for improvement in handling equipment in this type of mill than any other, as a vast amount of work is still done by hand.

60 Material is moved by hand to and from furnaces, to and from shears, and through the finishing processes of sheets, to an extent which should be eliminated by roller tables, chain conveyors, or even overhead monorail trolleys, any one of which properly installed would lighten the load now placed on human shoulders.

61 It might be said that the tube mill is a continuation of the plate or skelp mill, where tubes are made by bending up and lap welding the plate or drawing the smaller-sized skelp through a bell-shaped die to be butt-welded. Material-handling machinery for this type of work is well developed, and through the use of overhead cranes, conveyors, special pushers, etc., the manual handling of material is almost entirely avoided.

FORGINGS

62 Heavy forging is another branch of the steel industry that requires special handling machinery. It is a well-known fact that whenever steel is used to withstand heavy stresses, such as for battleship propeller shafts, heavy guns, etc., it is necessary to forge it. This is done under large hydraulic presses or steam hammers and it is necessary to handle the material rapidly after it is heated, to forge it as near as possible to the shape required before it cools. Fig. 14 shows a manipulator which when controlled by a skillful operator can handle a 3-ton forging with as great dexterity as a blacksmith handles a horseshoe. Such a machine may also be used to feed material into rolling mills of different types. Primarily, it is operated by three motors. The first controls the vertical travel of the end of the peel, the second rotates it on its own axis and the third moves it laterally by turning the car body on the truck. A handwheel controls the height of the rear or pivotal end of the ram so that it can be kept as nearly horizontal as possible. Two sets of springs are provided to give flexibility in both a vertical and horizontal direction and the tongs

are operated with compressed air furnished by a compressor mounted on the frame. The whole machine is moved on its track by two motors on the truck.

MISCELLANEOUS EQUIPMENT

63 Many types of conveyors are used in steel-plant auxiliary equipment, such as gas-producing plants, power plants, coal-pulverizing units, etc.

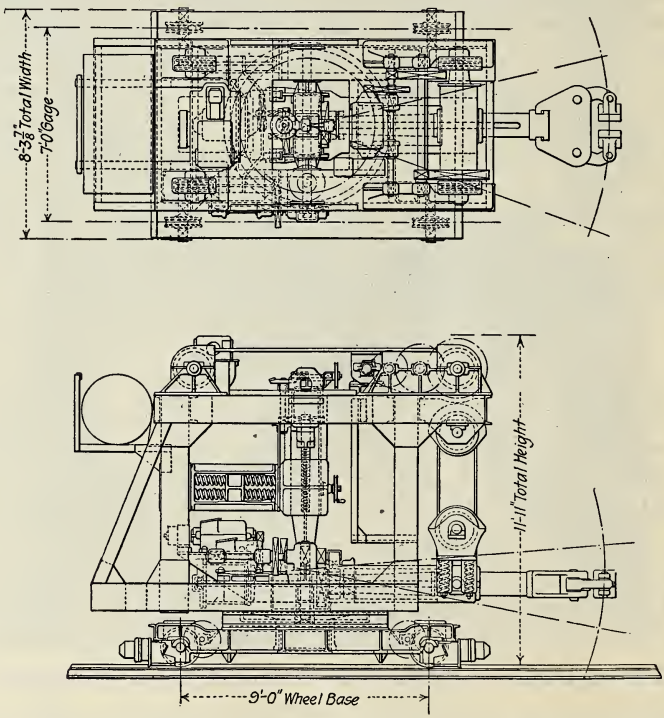

Fig. 14 Alliance Three-Ton Forging Manipulator

64 Among the most novel of these is one type of modern plant for pulverizing coal, in which the pulverized coal suspended in air is conveyed about the plant in pipes. This system has been successfully installed in several large steel plants where the pulverized coal has been used for open-hearth furnaces, reheating furnaces and boilers.

GENERAL

65 To summarize the problem of material handling in the iron and steel industry, it is quite apparent that there are many gaps in handling material from the ore mines to the finished steel which can be improved even with the present methods of manufacturing. Notably among these are the methods of conveying raw materials to the open-hearth furnace, and the many steps required from the blast furnace to the finished steel ingot. In other words they include the rehandling of material several times to obtain the desired refinement of steel, which is especially important where most of the material is handled in a molten condition with its consequent high cost. It is the writer's opinion that this refining process will change materially in the next few years through the medium of electricity, and as central power plants become more highly developed and more of our natural resources are put into use to obtain cheap electrical power, furnaces will be developed that will operate by a continuous process. They should be able to take the ore at one end of the furnaces, and with the addition of material at different points obtain the alloy desired at the pouring end thereby eliminating the handling now required from one furnace to the other by miscellaneous containers, ladles, etc.

66 In reducing ingots to the finished steel product, improvements should be made which will increase the safety of working conditions and eliminate as much as possible by proper ventilation at the mills and well-ventilated operating pulpits located at a distance from hot work, the disagreeable conditions arising from heat. Machinery should also be developed, wherever possible, to eliminate the necessity of handling hot steel with tongs around bar mills, rod mills, sheet mills, etc.

67 There is without a doubt a vast field of research, and improvement in this industry, as there is in all others, and it is for us, as engineers, to continue our endeavors and bring out these improvements.

DISCUSSION

In answering questions raised in the discussion, F. L. Estep, who presented the paper, said that he doubted if a machine could be invented to open the packs, since this is a problem of 75 per cent rolling and 25 per cent heat and the human element was needed. In a steel plant, he said, the wages were on a tonnage basis and therefore all equipment should be designed and constructed to meet the practical requirements of the workers and not to meet theoretical conditions. Handling and moving was a large item in the cost of manufacturing steel and steps were being taken to reduce it.

Stafford Montgomery asked about economical methods for conveying and storing large quantities of many sizes of light tubes in the finished condition between drawing operations, also whether steam locomotives could be economically eliminated from the yard proper of a steel company and electrical power used. In reply Mr. Estep stated that three or four methods could be applied in the handling of tubes between departments and storage between operations. A tube mill, as a rule, did not have overhead cranes in a big building but monorails could be installed which would quickly carry a good tonnage of hot or cold material. Or if it was cold tonnage, it could be handled with a magnet and a monorail. In regard to the question of steam against electricity, he said that the overhead-trolley system, nine times out of ten, would be too expensive. Difficulties in the way of maintaining and safeguarding would offset the saving in operating economies.

E. L. Shaner was of the opinion that the handling equipment in iron and steel plants could be divided into two classes, the largest of which was comprised of heavy and intricate machinery that required the attention of engineers who specialized in that equipment. On the other hand, there were many operations which required simple mechanisms and hand labor, and the reduction of handling costs on these operations was of great interest to iron, steel, electrical and metallurgical engineers.

No. 1841

THE ACCURACY OF BOILER TESTS

By Alfred Cotton, St. Louis, Mo.
Member of the Society

This paper points out the unavoidable inaccuracies involved in reports of boiler tests and the absurdity of assigning values carried out to the one-hundredth of one per cent to items which cannot possibly be measured so closely.

The various factors which enter into boiler-test computations are taken up one by one and errors which may be made are covered and the best methods of reducing them considered.

The most important elements are of course the determinations of water and coal used, and of steam produced, both as to quality and quantity. It is shown how the error in obtaining the B.t.u. in the coal burned may easily reach 2.5 per cent in either direction, the water evaporated may be at least one per cent out of the way, and hence the efficiency reported may be 3.6 per cent too high or low.

A LARGE proportion of boiler-test reports are presented with heat-balance figures carried to one-tenth and even one-hundredth of one per cent. Those who have had much experience in boiler testing know that such apparent accuracy is impossible at present and therefore misleading. In a heat balance recently published the percentage portion does not add to an even hundred. Since the last item is simply the difference between what is known and 100, why not make it agree? The deviation from 100 makes it appear that the items are very precise indeed.

2 How accurate is it possible to make a boiler test? To determine this, every item entering into it must be considered.

3 First consider the measurement of the water. If the water is carefully weighed or measured by suitable meters, the amount fed may be ascertained well within plus or minus 1 per cent. If meters are used, they may be calibrated before the test, and perhaps afterward as well, to insure this accuracy. Proper precautions

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may be taken to prevent leaks from blow-off cocks, in feedwater heaters, and so forth. Care can also be taken to see that the condition of the water in the gage connections is the same at the end as at the beginning of the test. If this is not done, error may be introduced to the extent of 1 or 2 in. due to the difference in the weight of cold water in the gage connections before blowing off, and of hot water afterward. With an eight-hour test run at rating, the error may be more than 1 per cent from this cause.

QUALITY OF STEAM

4 Then there is the quality of the steam to be accounted for. The entrained water in the sample can be ascertained very closely, but it is sometimes stated with much greater precision than the observations warrant, or than is necessary. In a recent boiler-test report the proportion of water in the steam was given to one-ten thousandth of one per cent!

5 The temperature of the steam in the expansion chamber of a throttling calorimeter need only be read to the nearest degree; and when averaging the readings, the result need not be expressed beyond one-tenth of a degree. A glance at a steam table will show that the variation is so slight that carrying it further is futile.

STEAM SAMPLING

6 But does the sample of steam so examined represent the bulk steam? That is something which is not known, and the following discussion seems to show that it is *not even nearly representative.*

7 The design and position of the sampling tube are matters of some contention. It is accepted practice to perforate the sampling tube and place it horizontally across the pipe. It is probable that entrained water is commonly in such small particles that it is easily carried in suspension at usual steam velocities. Water flowing along the bottom of the pipe is likely to be mostly condensed steam, and is therefore rightly avoided by placing the sampling pipe horizontally.

8 The A. S. M. E. sampling tube is made of ½-in. standard pipe and is provided with twenty ⅛-in. holes arranged in "irregular or spiral rows," and usually in a more or less regular helix. It should be inserted in the steam main "at a point where the en-

trained moisture is likely to be most thoroughly mixed." The word " likely " is highly commendable.

9 It is most probable that the design of a steam-sampling device should be somewhat similar to that of a rain gage; and if this is the case, the A. S. M. E. tube is as unsuitable for its purpose as anything which could well be contrived. One of the fundamental requirements in the location of a rain gage is to avoid the proximity of objects which might shelter the instrument or cause eddies. This requirement is ludicrously violated by the downstream holes in this steam-sampling tube. It would appear that the best position for the holes is directly upstream, and that all holes should be in a line facing in that direction instead of being distributed around the tube. For the smaller steam pipes, a smaller number of large holes could be used, placing them at least, say, 3 diameters center to center, and making their total area about equal to the twenty $\frac{1}{8}$-in. holes or, say, 0.25 sq. in.

10 The most accurate results would be attained with such upstream holes if the velocity of the steam entering the holes approximated that of the bulk flow along the steam pipe. Then the steam and water particles would enter the holes with the least disturbance of their general direction, and consequently the sample would be most representative.

11 The calorimeter nozzle is $\frac{5}{64}$ in. in diameter. The higher the pressure, the greater the weight of steam flowing through it; but owing to the greater density, the volume entering the sampling holes does not vary much. The velocity of the steam entering each of the twenty $\frac{1}{8}$-in. sampling holes is from 1100 to 1200 ft. per min. Steam-pipe velocities run from 5000 to 14,000 ft. per min., and modern tendency is toward high steam velocities. Therefore the sample would generally be taken under the conditions shown in Fig. 1. The bulk steam flows along the pipe much more rapidly than the sampled steam flows into the hole. The sample is richer in water than the bulk steam, because most or all of the water in the thin cylinder of steam directed thereat enters the hole owing to its inertia, while a considerable proportion of the steam does not enter but is deflected around the sampling tube. If the bulk steam is flowing much more slowly than the sample flows into the hole, then all the steam in the thin cylinder will enter the hole together with some outside of this thin cylinder, and the water belonging to this outer portion will not be included in the sample, but will strike the surface of the tube and flow around it as in Fig. 2. In

illustrating these conditions the dots represent water and the solid arrows, steam.

12 The faster the bulk steam flows in the steam pipe the more water the sample will contain. If its velocity is twice that of the sample entering the hole, then the area of the thin cylinder supplying the hole will be one-half that of the hole. The water entering the hole will be that contained in the area of the hole, or twice what it should be, and the percentage of moisture as observed should be multiplied by 0.5.

13 It will now be shown that holes in any other position around the sampling tube than the directly upstream position just discussed, are highly objectionable. As the steam flows past the

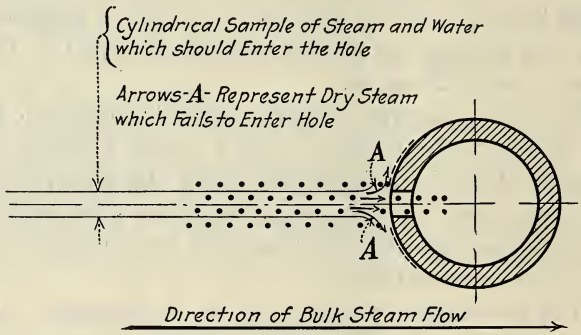

Fig. 1 Bulk-Steam Velocity Greater Than Sample

sampling tube, it is deflected and slows down into eddies on the downstream side. But, as is well known, the water particles are heavier and tend to travel in straight lines. Therefore those in line strike and wet the tube, and then flow around it, adhering to the surface, to the downstream side. Choosing a single downstream hole, its watershed will be the outside diameter of the tube multiplied by the diameter of one of the $\frac{1}{8}$-in. holes. Standard $\frac{1}{2}$-in. pipe is 0.84 in. in outside diameter, so that the collecting area is 0.105 sq. in. The area of the $\frac{1}{8}$-in. hole is 0.0123 sq. in., so that a hole pointing directly downstream receives over eight times its share of water. This is probably not quite true because a little of the adhering water may be picked off again by the steam flowing over it. Similarly, the watershed of a hole pointing at right angles to the stream will be one-half of the above area, and the hole will receive four times its share. Suppose that holes are drilled at regular 90-

deg. intervals around the tube, including directly upstream and downstream holes. Then the water-collecting power of four such holes is as follows:

Upstream	1	times actual
Normal	4	" "
Downstream	8	" "
Normal	4	" "

and the average collecting power is 4.25 or, say, 4 times the true proportion.

14 It may be contended that instead of the water particles traveling in straight lines, some will be carried into or away from

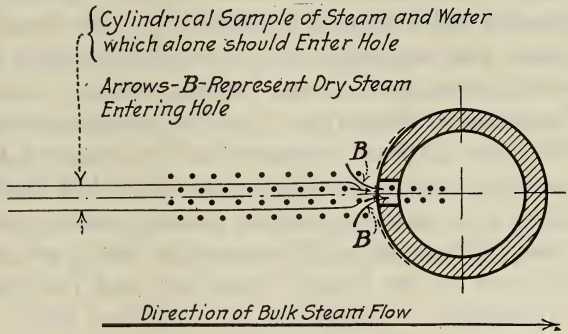

Fig. 2 Bulk-Steam Velocity Less Than Sample

the sampling hole, and that more or less than the receipts of the watersheds will enter the appropriate holes. It may also be reasonably contended that with high steam velocities much steam will enter the tube through the upstream holes and leave through the downstream ones. Thus the water adhering to the outside of the tube would not enter the downstream holes as suggested. On the other hand, it is probable that much of the water which came in with this transient steam would be deposited on the inner walls of the tube, and the steam leaving the downstream holes would be drier than when it entered. Whatever it left behind would enrich the steam sample with water. All these things are most readily admitted. But it seems reasonable to think that the actual water present in the steam may be very much less than that observed when we use the A. S. M. E. tube.

15 If there is much priming, the error in determining the

amount of water carried over may approach plus or minus 5 per cent of that evaporated. But assuming orderly operation and careful observation, the accuracy of the report on water evaporated may be from plus or minus $1\frac{1}{2}$ to 2 per cent; but it cannot well be guaranteed to be closer.

ESTIMATING COAL CONSUMPTION

16 The next item to be considered is the coal, and this may easily be weighed well within plus or minus 1 per cent. However, this is but the coal fed to the furnace, and to know the quantity actually burned in relation to the water evaporated, we must be sure that there is the same amount of fuel in the furnace and in the same stage of combustion at the end of the trial as at the start. The only way to find this out is to look at it. Only those who have done this know how impossible it is to accomplish much more than make a careful guess, and it must be done quickly because conditions are changing momentarily. It is not uncommon for different observers to vary in their estimates to the extent of 10 or even 20 lb. of coal per sq. ft. of grate area. In a test during which a total of 250 lb. of coal was burned on each square foot of grate, the error from this source might amount to over 5 per cent. The error will depend on the length of the test, and for 12-hour tests may very reasonably be placed at not less than plus or minus 1 per cent.

17 But, unfortunately, this coal weight does not mean anything of itself. We must at least know the weight of *dry* coal; and if the moisture present in the coal is much over 2 per cent, allowance should be made for the heat used to evaporate and superheat it. To find out how much moisture there is in the coal, we must have a representative sample, and then we must analyze it for moisture *while it is representative.* While working down a laboratory sample, the coal will either gain or lose moisture. In a warm, drafty room and especially on a warm floor, it will generally lose moisture, perhaps as much as the equivalent of 3 per cent of the coal or even more. While it is properly bottled up, no further change will occur; but at the laboratory, while being finely ground and while weighing samples, further change will undoubtedly occur and this may be either in increase or reduction of moisture. During all these processes of sampling from the coal pile to the laboratory drying oven it does not seem possible to certify the moisture closer

than plus or minus 1 per cent, and it may easily be out plus or minus 2 per cent. Taking the alternative method of drying the sample of coal during the test, there is the difficulty of securing a representative sample, for we are now sampling coal which has not been weighed to the firing floor or hoppers; and there is also the further difficulty of crude drying and weighing apparatus. However, this method is likely to be a little more accurate than the orthodox one, and with reasonable care in selecting samples, results may be obtained within plus or minus 1 per cent.

18 On examining the report of any boiler trial, the first thing looked for is the efficiency at the trial load. Without this, the report does not generally convey information of very much value. To get the efficiency we must know the heating value of the coal; and whether the heating value is found by calculation from the analysis or by the calorimeter, the essential feature is the extent to which the sample is representative. To fully satisfy this requirement its particles should have come from widely distributed points in the original bulk coal used during the trial. Of course, this condition is very far from being reached even with the orthodox method of quartering down, unless the test is with powdered coal. Still, such a sample is reasonably representative if the work of getting it has been conscientiously done.

SUMMARY OF ERRORS

19 It is well here to summarize the possible errors and see where they may lead, and this has been done in the following tabulation, keeping them even lower than suggested.

COAL	ERROR
Weighing	± 0.5 per cent
Estimating amount of fuel in furnace at start and stop	± 0.5 per cent
Total error of weighed coal	± 1.0 per cent
Variation between original percentage of moisture and that in laboratory sample plus failure of sample to truly represent bulk	± 1.0 per cent
Failure of heating-value sample to represent bulk.	± 0.5 per cent
Total error in analysis	± 1.5 per cent
Total error in B.t.u. in coal (weight and analysis) on which efficiency is based	± 2.5 per cent

WATER	ERROR
Weighing or metering, starting and stopping test, gage glass, leaks	± 0.5 per cent
Failure of sample of steam to truly represent bulk steam as to entrained water	± 0.5 per cent
Total error in water evaporated	1.0 per cent

B. t. u. in coal burned may vary from 97.5 to 102.5 per cent.
Water evaporated may vary from 99.0 to 101.0 per cent.
Reported efficiency may vary from 99.0/102.5 to 101.0/97.5, or 96.6 to 103.6.

20 Therefore if the efficiency really attained in any boiler trial is 78 per cent, the report may show as low as 75.4 per cent or as high as 80.8 per cent if all the errors happen to be in one direction. It may be thought that this possible error is placed at a higher figure than occurs in practice; but it must be remembered that generally the errors will be in different directions and so tend to neutralize each other, and therefore it will be seldom that they are all one way. Consequently if a sufficiently large number of trials are made on the same boiler under the same conditions, some of the tests will nearly always be found to deviate from the average quite as much as the total possible error here suggested.

21 If the coal and water deductions are relied on, we know the efficiency and the load. But in view of what has been said it is very doubtful if the regular carefully conducted boiler test burning hand- or stoker-fired coal can be guaranteed to be closer than within plus or minus 3 per cent.

THE HEAT BALANCE

22 The mere presence of a heat balance is often considered as a final guarantee as to the correctness of the test. A boiler-test report without a heat balance is just as good as the other kind as far as efficiency at load is concerned. But the heat balance confers dignity and precision on the test report because all the items add up to 100 and this seems to prove the correctness of everything concerned in it. However, since it does not " balance " or rather that we *make* it balance, the term " heat balance " is not without humor. If accountants submitted items in their balance sheets called " unaccounted for " amounting to $5 in every hundred dollars there would be consider-

able unpleasantness. Of course, a heat balance is different because there is the excuse that no way has been discovered to find out the amounts of some of the items and the others are not as accurate as is generally believed. In a case of four consecutive tests made on the same boiler by an engineering professor of deserved repute and large boiler-testing experience, the "unaccounted for" was respectively 8.7, 0.4, 6.4, and 7.4 per cent. The trials run by Dr. Jacobus on the Delray boilers were among the most careful ever made, and the "unaccounted for" item runs from 1.51 to 4.83 — a difference of over 3 per cent. Incidentally, the efficiencies in the series of tests vary over plus or minus 1 per cent from their average rating-efficiency curve.

23 The real value of a heat balance lies in the presentation of the various losses and their amounts, and therefore it is more a *statement* than a *balance*.

24 Care has to be taken to get an average sample of flue gas. Its composition sometimes varies greatly across the width of the setting — as much as 5 per cent of CO_2 according to some experiments, though perhaps some of this variation was caused by the quality of the gases changing rapidly between taking samples. It seems likely that if the gas sample is drawn from some point well within the gas flow, it is fairly representative, for if that were not so, the "unaccounted for" would show a still larger variation than it actually does. This also applies to exit gas temperature, where the radiation effect may affect the readings 50 deg. fahr. With care in placing the instruments the error is perhaps within plus or minus 20 deg. fahr., or about plus or minus 1 in the percentage of heat loss to the stack.

EFFICIENCIES

25 Even the efficiency as now reported does not provide a proper basis for making comparison between different trials under different conditions. An efficiency of 75 per cent with a steam pressure of 225 lb. is a better performance than the same efficiency at 75 lb. pressure. To be serviceable for comparison, the efficiency should be based on what is theoretically possible in each individual case, and this is governed by the temperature of what is being heated and not by the temperature of the atmosphere.

26 At first sight, it appears that it is not theoretically possible to get the exit gas temperature below that of the boiling point. Therefore, the efficiency should be based on perfect com-

bustion with the theoretical amount of air and the products of combustion cooled to the boiling point for the existing pressure. This is not very difficult, and the 100 per cent figure would be set for the particular fuel and steam pressure under consideration. The efficiency would be the percentage of the figure attained on the test and might be called the " actual " efficiency to distinguish it from the efficiency as now reported, which might be called the " theoretical " efficiency.

27 Whether the mean temperature of the water in the boiler or that at the gas exit should be used, is another consideration.

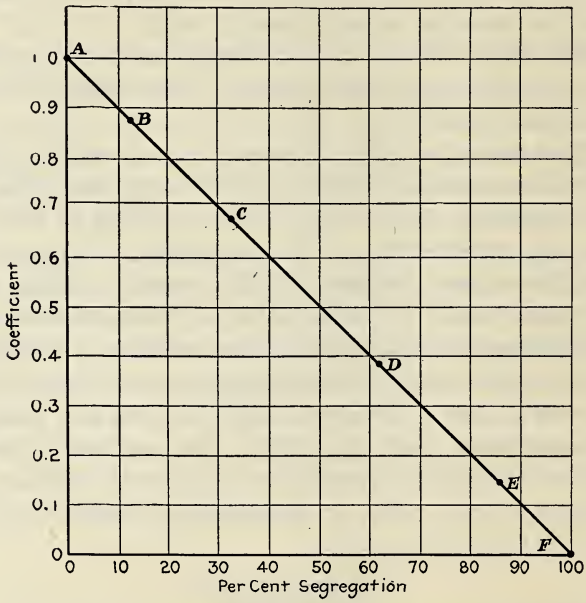

Fig. 3 Effect of Segregation of Feedwater

In a horizontal water-tube boiler, for instance, the lowest tubes may be 15 feet below the water level. The added pressure due to this head of water is 6 lb., making a total pressure of 156 lb. when the gage pressure is 150 lb. This increases the boiling point 3 deg., and either the water is hotter at the bottom of the boiler or no steam is made in these tubes.

28 The temperature to be used is really dependent upon the extent to which the cold feedwater is segregated. The boiling point may be considered as the proper temperature to use with a

boiler in which the feed is so mixed with the circulation that no cold water comes in contact with the heating surface, while with a separate economizer or a flash boiler, the temperature of the feedwater should be adopted. But difficulties at once arise with boilers where there is partial segregation. In boilers having integral economizers the feedwater may be wholly or partly segregated. In integral economizers of the Badenhausen type, for instance, there is considerable circulation within the economizer, and the temperature of the water where the gases leave will certainly be well above that of the entering water. A somewhat similar condition obtains in the rear bank of the Stirling boiler which is, or used to be, often called the "economizer bank."

29 It appears, then, that the design of boiler must be taken into consideration and the temperature chosen be that of the boiling point, or of the cold feedwater, or somewhere between these points according to the extent of segregation of the incoming feed. A chart like Fig. 3 could be used and a position on the curve agreed upon for every type of boiler, such as the following:

POSITION	TYPE OF BOILER
A	Horizontal Water-Tube
	Scotch Marine
	Horizontal Return Tubular
B	Stirling with rear circulators
C	Stirling without rear circulators
D	Boilers with integral economizers having free circulation
E	Boilers with integral economizers having restricted circulation
F	Boilers with separate economizers
	Flash boilers

It is not intended to propose that the positions thus suggested by way of illustration be accepted, either absolutely or relatively. But as an example, with a boiler having 25 per cent segregation, the coefficient as read from the chart is 0.75. Suppose the boiling point is 370 deg. and the feedwater enters at 130 deg., then the difference is 240 deg. and 0.75 of this is 180 deg. Then $180 + 130 = 310$ deg. fahr., which is the temperature to be used.

30 It might be advisable to make some kind of an allowance for the rate of driving, because this would have some effect on the temperature to be used in those boilers having partial segregation of the feedwater.

31 Probably the simplest way to dispose of the matter is to take the temperature of the water in the boiler at the point where

the gases leave, making regular observations during the test. In this way there would be no contention whatever as to the proper temperature to use. The average of these observations would be recorded in the test report.

32 The efficiency based on the temperature of the water in the boiler is the only efficiency by which different boiler performances may be intelligently compared. But it must be remembered that the limit of temperature on which it is based is artificially imposed. It may be considerably lowered by the installation of an economizer, and perhaps still more so by apparatus yet to be devised or more generally adopted. When an economizer is added it may be that the efficiency — which will now be based on the feedwater temperature — is lower than before. Of course, this is quite proper because this is really the actual efficiency of the apparatus as a heat generator and absorber. But for the comparison of one steam plant with another or for changes in the circumstances of the same plant such as the addition of the economizer, the efficiency based on the heat value of the coal must be used. Therefore it appears highly necessary that both of these efficiencies be reported in every test. These efficiencies may be stated as:

$$\frac{\text{B.t.u. in steam}}{\text{B.t.u. in dry coal}} = \text{Theoretical Boiler and Furnace Efficiency}$$

and

$$\frac{\text{B.t.u. in steam}}{\text{B.t.u. available}} = \text{Actual Boiler and Furnace Efficiency.}$$

33 All comparisons of efficiency are properly based on the load in percentage of the boiler rating. When a superheater forms an integral part of the boiler, the heating surface is greatly increased. The efficiency is greater owing to the increased heating surface reducing the temperature of the escaping gases. But still the efficiency is plotted against boiler rating — in other words, against boiler heating surface only. This is wrong. The combination should be rated at some higher horsepower than that of the boiler only. Superheating surface should be rated as the equivalent of so much boiler heating surface, though it would be a very difficult matter to decide just how much to allow, as its heat-transfer rate is quite different from that of boiler heating surface.

34 The same thing applies to economizers. What is the proper way to report a boiler test where there is an economizer? I quote from a test report of unquestionable authority:

Water-heating surface	4400 sq. ft.
Builder's rating	440 hp.
Economizer heating surface	6290 sq. ft.
Hp. developed for boiler	1231
Hp. developed for boiler and economizer	1350
Percentage of rating developed for boiler	277
Percentage of rating developed for boiler and economizer	305

What is the meaning of the last item? The one above it is the regular statement of percentage of rating. But the boiler did not develop more horsepower or a higher percentage of rating because there was an economizer near it. The economizer did this extra work — not the boiler. What percentage of the *rating of the economizer* did the economizer develop?

35 The steam, or its equivalent when electric motors are used, consumed in driving stokers and mechanical-draft apparatus should always be ascertained and reported as a percentage so that it may easily be deducted from the efficiency by any one sufficiently interested. In testing all or most other apparatus, the efficiency is output divided by input; but in boiler testing we do not know yet how to accurately measure the output. We weigh the water going in and boldly state it as output. Surely the available steam sent away from the boiler is the real output, and if so, all steam used to make the boiler and its accessories work should be allowed for. As conscientious engineers, we have always strictly seen to it — and shall continue to do so — that such things as internal-combustion engines, hot-air engines and the like, shall be tested with a brake so that we know exactly what is their actual output; and the more power they consume in making themselves work, the worse for them. But we seem to prefer to avoid telling any one what the real output of a boiler is.

36 Economizers raise the efficiency by lowering the flue-gas temperature, and they sometimes do this to a point where the gas temperature is too low to produce sufficient chimney draft, in addition to increasing the draft resistance. So an induced-draft fan is used, but the power taken to drive it is rarely considered.

37 The following very imposing array of data is sometimes presented:

Water evaporated, actual, per lb. of coal as fired
Water evaporated, actual, per lb. of dry coal
Water evaporated, from and at 212 deg., per lb. of coal as fired
Water evaporated, from and at 212 deg., per lb. of dry coal
Water evaporated, from and at 212 deg., per lb. of combustible

These items are sometimes carried to one-thousandth of a pound and none of them means anything. Any kind of evaporation per pound of coal in any condition is without meaning unless the heating value of the coal is known, and knowing this the efficiency, which is what is desired, can be easily determined.

38 In *The Engineer* (London) of February 25, 1921, H. V. Whittaker suggests basing the evaporation on "standard coal" and he further suggests giving this theoretical coal a value of 12,600 B.t.u. If all tests were reported on the basis of this standard coal we would have a definite standard for the comparison of different performances. But why not go a step further and drop the meaningless evaporation per pound of coal altogether and report the result as B.t.u. in the steam per 100 B.t.u. in the coal? This figure is the efficiency without further ado. There is nothing difficult about it. In calculating the test, the B.t.u. in one pound of liquid is subtracted from the B.t.u. in one pound of steam, superheated or otherwise, and multiplied by the weight of water evaporated. The "factor of evaporation" disappears, and the correction for moisture in the steam is just as easy. The B.t.u. per pound of coal is multiplied by the coal weight and divided into the B.t.u. taken up by the steam. The quotient is readily expressed as B.t.u. in the steam per 100 B.t.u. in the coal. The B.t.u. per hour taken up by the steam is easily converted into the obsolescent "boiler horsepower."

39 The object of all boiler tests is comparison, either with some other test or group of tests, or with our general experience, which is the same thing. The report should therefore contain everything which will show the relative arduousness of the conditions under which the test was made, including the variation in load.

TOLERANCES

40 In all manufacturing operations it is realized that nothing can be made absolutely correct. Therefore tolerances are allowed. Whether the suggested plus or minus 3 per cent for boiler trials

is considered too large or not, the fact remains that the performance of guarantees is demanded without any tolerance whatever. It is said that if guarantees should carry a tolerance of, say, minus 2 per cent, manufacturers would raise their guarantees by 2 per cent, relying on this margin. If all manufacturers followed this policy they would be on an equal footing and no commercial harm would be done. As matters stand now, however, the prudent manufacturer virtually sets his own tolerance by guaranteeing something less than he knows he can accomplish; while if the proposed tolerance is adopted he can guarantee all he knows he can do, and let the tolerance take care of the testing errors.

41 There is nothing whatever to be gained by reporting efficiency more closely than the nearest tenth of one per cent, or even the nearest half. Not only is such apparent precision as 1 in 1000 unwarranted by the exigencies of boiler testing; but engineering judgment is quite unable to make use of it. It is mentally impossible to differentiate between the relative values of 79.9 and 80.1 per cent efficiency and it seems only sensible to call it 80 per cent. Especially is such overprecision absurd, inasmuch as it is not certain that it was not really 79 or 81 per cent, or even a little further away.

DISCUSSION

Grant D. Bradshaw. The paper covers a condition which is very imperfectly understood not only by the layman, but also by a great many engineers as well. A boiler test, by many people, is looked upon as the "last word" in accuracy, and the author very properly brings out the various unavoidable errors which may creep in, and which, in the aggregate, amount to a very considerable figure, if it so happens that they should not tend to compensate one another.

The author's statement that "it is very doubtful if the regular carefully conducted boiler test can be guaranteed to be closer than within plus or minus 3 per cent" is probably correct. In considering this matter, however, it must be borne in mind that under usual conditions a number of the possible errors will counterbalance each other. This seems to be substantiated by the results of Dr. Jacobus'

tests on the Delray boilers, where the unaccounted-for item runs from 1.51 to 4.83 per cent.

With this in mind, it is evident that the figure of radiation and unaccounted-for losses, which is shown up by the heat balance, becomes of considerable importance, in judging the probable value of the results. Radiation and conduction losses in the ordinary boiler are practically constant, and very seldom run over 5 per cent. When, therefore, a boiler test is submitted which shows radiation and unaccounted-for losses as high as 17 per cent, which is a case that has actually come under the writer's observation, it becomes immediately evident that there is something radically wrong with the essential measurements.

The problem of sampling steam to determine the amount of moisture contained in it is very closely allied in principle to the sampling of blast-furnace gas to determine its dust content. The dust determination tests are well standardized at present. In collecting the gas sample a Pitot tube is used with the opening pointing upstream. Through this tube is passed gas at a velocity exactly equal to the velocity of the gas in the main from which the sample is being drawn. It has been well determined that any variation in the velocity of sampling, as compared to the velocity of the gas in the pipe itself produces erroneous results. The sampling of moisture carried in steam is a similar problem, and the author's remarks on the matter cover the case very thoroughly. The writer cannot agree entirely with some of the details of the author's theory regarding whether the amount of moisture will be increased or decreased in given cases, but the general premise is certainly correct, and should be taken into account in making such a test.

E. H. Tenney. Although it is true that the possibilities of error in conducting a boiler trial are within the limits of accuracy, as set forth by the author, the author does not point out a way or method for the elimination of such error. Experience has proven that a boiler test conducted by reputable engineers, in accordance with the present standards as laid down by the Society, gives results which can be relied upon and which are accurate for comparison with other trials and conditions. It is very probable that if quantities were not figured as accurately as is generally the case the error in boiler efficiency would be much more than that expressed by the author of this paper, and for this reason we have found that by carrying the calculations to the number of decimal places in keeping

with the weight that they bear to the boiler trial, more accurate results will be obtained than if the second decimal place and sometimes the first is dropped indiscriminately.

It appears as if the boiler trials under discussion by the author are the ordinary trials conducted in a haphazard method by certain manufacturers of so-called fuel-saving equipment, and others interested primarily in confirming a suspicion as to efficiency which they had previously determined. The writer believes it to be the consensus of opinion among engineers that boiler trials should not be compared or their results used unless all conditions surrounding the test are also reported. Too many tests are reported for publication wherein the length of duration is not stated or the duration is over such a short period of time that errors in observation appreciably affect the efficiency reported. Boiler trials, in order to be as accurate as possible, even when following the standard code of testing, should be carried over a period of at least twenty-four hours. On one of our recent tests the total coal burned per hour amounted to 6 tons or 144 tons during the period of twenty-four hours. The coal was accurately weighed on scales which were calibrated and checked at regular intervals throughout the test and it is hard to conceive of an error of 1 per cent either plus or minus as suggested by the author. The error in this case would amount to plus or minus 2800 pounds of coal. The same applies to the weighing of water, with this difference, however, that a greater amount of water is weighed and an error of 1 per cent over a twenty-four-hour test would be a very appreciable amount.

Although it is true that a heat balance is made to balance, it serves as a check on the reported results of any test. At certain efficiencies, certain losses will be present, and even though the balance is 100 per cent, the unaccounted-for items are relatively small compared to the boiler efficiency and the other important losses.

The author makes the statement, that under the present method of reporting boiler efficiencies there is not a proper basis for making comparisons. The term "from and at 212 deg." was devised for the express purpose of making all reported boiler trials comparable and even though the author suggests the reporting of B.t.u. per pound of steam per 100 B.t.u. in the coal, the answer is exactly the same as that derived from our present conception of boiler efficiency, "from and at 212 deg."

John E. Bell. Mr. Cotton has discussed many different kinds of errors that can enter a report of a boiler test. That errors are made of the magnitude that he mentions, or even greater magnitude, the writer believes there is no question. The best tests, of course, are those in which these errors are reduced to a minimum, which requires a degree of judgment and skill on the part of the engineer in charge that is possessed by very few. As a consequence of this, it has been the writer's experience that very few boiler tests have been made in this country which are accepted unreservedly, and from what he knows of the method of making boiler tests abroad, a still greater uncertainty exists there. Assuming that every effort has been made to attain accuracy, it still remains a matter of judgment as to the correctness of the estimate of combustible in the furnace at the beginning and end of the test and on the fairness of the coal sample. The mistake that must necessarily be made in estimating the combustible in the furnace is reduced as the length of the test is increased, and it is for this reason that little consideration is given to boiler tests that are not at least twenty-four hours long. There is no way, however, as far as the writer knows, in taking a coal sample to escape largely relying on the judgment of the engineer in charge of the test. Even granting that a fair sample of the coal has been taken, it will usually be found that different laboratories report different heat values and different moisture contents, and the difference in heat value will often vary the efficiency 2 per cent or more. The writer does not believe this is due to variation in the heat value between the samples submitted to the different laboratories as much as it is to differences in the way the sample is handled in the laboratory. In two instances it was found that laboratories that made a specialty of determining the heat value of coal samples uniformly reported results with a constant error in the determination. In one case the heat value was always about 2 per cent high, and it was found that this was due to the fact that in grinding the sample the grinding machine was not entirely cleaned out and what was left was principally bone. In the other case the heat value was always low due to defective insulation of the calorimeter. The engineers who have most to do with boiler tests seem to the writer to be the worst critics of the laboratory results.

After all is said and done, importance can be attached only to boiler and furnace efficiencies determined in a long series of tests, each test being of at least twenty-four hours' duration and checked

by one or more duplicate tests. In such a series of tests heat values are always determined by at least two laboratories and must check before the test is reported. Even tests of this sort are of no value unless every effort has been made to attain the very highest efficiency and to eliminate all sources of errors. The heat balance for such tests is used in determining the importance to be attached to the results. If the unaccounted for is between one and three per cent, it is obvious that the series of tests has been carefully made and that the loss due to smoke has been eliminated. A heat balance on an individual test does not have this same value for the reasons pointed out by the author, but nevertheless it is a good thing to have. If the unaccounted for is seven or eight per cent, the worth of the test is doubtful as either inexcusable errors have been made or else there was a large loss due to incomplete combustion. If the unaccounted for is very small, say less than one per cent, the test is also known to be unreliable. The hardest item to determine accurately in the heat balance probably is the carbon in the ash, and in many cases it is almost impossible to get this accurately. The heat balance again is of assistance among a series of tests in checking this.

The writer agrees with Mr. Cotton that results are sometimes carried to an absurd number of decimal places, but he doubts if anyone who knows enough about boiler test work to form an intelligent opinion is ever led to believe that the accuracy of the results is in any way indicated by the number of decimal places used. Sometimes calculators are required to run out their results to avoid errors that might otherwise creep in, due to their lack of judgment in dropping decimal places. Anyway, if a person desires to spend the additional time in long multiplications or divisions, he is going to do so regardless of what others may think of his wasted efforts. It is also true that the table of results in water evaporated, mentioned in Par. 37, is in most cases unnecessary, but the time spent in figuring the test is relatively of no importance and occasionally every one of these evaporations is needed in making comparison of different sets of tests. If an engineer wants to add to his test figures on "true efficiency" on "standard coal" or has some scheme for separating boiler and furnace efficiency, why not let him report it? There will never be any agreement amongst engineers regarding such results, and a calculation that one believes of value, another will consider worthless.

The uncertainty that exists in reporting tests on boilers with

individual economizers, to a smaller extent, exists also in connection with superheaters, and with superheaters the per cent of rating developed is figured both ways, and as long as the boilers are manufactured by one company and the economizers by another, the writer believes tests including economizers will be figured in this way. Granting that it may be somewhat unscientific, it nevertheless gives results in a form that is often referred to.

The writer believes that the paper is important in directing the attention of engineers to the unreliability of reported boiler tests, but he does not think that it should lead to any revision of the present code in reporting tests by the Society for the reason that engineers are going to report tests as they please, even though there is another amendment to the constitution to the contrary.

G. C. Vennum. Undoubtedly considerable time and energy are wasted in carrying statements of boiler test reports to points beyond the decimal point. A 100 per cent accuracy of all the items observed in boiler performance including fuel and measuring of water is recognized as being impracticable, and the deviation from accuracy as estimated by the author should represent a very good working range.

We do not agree with Mr. Cotton that the heat balance is something to be taken lightly. This is the best medium, as the writer sees it, for judging the accuracy of the report, and by which one can draw conclusions as to characteristics of performance.

The author proposes that a schedule be established for reckoning the performance of different boilers under various conditions, referring particularly to temperature of feedwater and type of boiler. The writer does not believe that any such adjustment would offer a good basis for comparison. What we want to know is the relation between what we have to start with as expressed in heat units in the fuel, and what we get in the transformation of energy in fuel to available energy for mechanical use, which with the steam boiler is the heat units in the steam. Boiler design, feedwater temperature, steam pressure, superheat, etc., are all factors in the result, and cannot be adjusted and give a true statement of efficiency.

G. R. Davison. The degree of reliance which can be safely placed upon the results of a boiler test depends upon a knowledge of the conditions under which the test was made. There are vast opportunities for large errors due to improper installation of measur-

ing apparatus, to lack of precautions for maintaining necessary conditions, to inexperienced judgment in taking observations, and to neglect in applying corrections for unavoidable errors in methods.

The paper has brought out the fact that a plus or minus three per cent or higher error in results may be possible in a carefully conducted test when all precautions are taken to make the results reliable. It is evident, therefore, that the value of the results of a boiler test depends upon the experience of the men who conducted it.

The author mentions a number of precautions that must be observed during a test, and calls attention to the importance of ending the test with the same conditions of water in the gage as existed at the beginning of the test. In this connection it is of equal importance to have the rate of steaming, rate of feeding, and the conditions of combustion and of fuel bed, of draft, temperature, and pressure, also the same at the beginning and end of the run.

Quality of steam is a difficult factor to determine accurately. The calorimeter is not a perfect device, and a good deal of care must be exercised in the placement and installation. When employed with a sampling tube, such as recommended by The American Society of Mechanical Engineers, it has been found that more accurate results have been obtained by using a single line of holes on the upstream side. The able discussion of the author upon the bearing of the location of the holes in the sampling tube indicates that something might be done in a practical investigation of the subject. It would seem that the arguments against the line of holes on the downstream side might be met by constructing a tube with a number of saw cuts made at intervals across the tube on the downstream side, to replace all holes upstream and downstream. The area of the water shed would then equal the area of the hole, and using the arguments presented, all of the water except what is picked off and carried on should enter the saw cuts.

The author calls attention to the difficulty of estimating the condition of the fire at the beginning and end of the test and indicates the value of making a long test to reduce the errors from this source. In the same manner all other errors due to conditions at start and end of tests may become appreciable in a test of short duration.

Under the term "actual efficiency" which the author uses, the principal difficulty is in uniting upon a standard conception for the temperature of the water. Without such a conception a great deal of confusion would result in making any comparison on this basis.

The proposed use of charts for the purpose of accounting for the extent of segregation of the incoming feed introduces factors which may be subject to a variety of interpretations. Furthermore, it appears to lead to a refinement such as the author has been trying to get away from.

In considering the question of arriving at a figure for boiler output, leakage through handholes cannot well be accounted for, as there is no means for measuring it. However, actuation and fan power can be measured accurately by means of flow meters which are accurate to two per cent and as this power requires not far from two per cent of the total coal consumption, the error from the use of the meters for this purpose is negligible in the net results.

Flow meters may also be used to advantage in measuring the boiler output direct and providing a check on the results.

In the question of tolerance it would seem wise in cases where the manufacturer is concerned, to come to an agreement upon this subject with him before the test. It will then be a question of conducting the test in such a manner as to come within prescribed limits.

C. F. Hirshfeld, C. Harold Berry, Paul W. Thompson and W. A. Carter. The author's emphasis on the consideration of attainable accuracy touches a vital point in engineering practice, and most engineers will doubtless agree that much attention should be given to this important subject. Too many people think that engineering tests are accurate to a small fraction of one per cent, and too many testing engineers, either willfully or thoughtlessly, give a false impression of high accuracy by presenting an imposing array of results computed to an absurd number of significant figures. Too many lawsuits follow from such false conceptions.

A clear distinction must be made between various types of boiler tests. On the one hand we have commercial tests, and on the other hand we have research tests. In a careful research, no pains may be spared to secure the highest attainable accuracy, whereas in a commercial test it often happens that a low degree of absolute accuracy is ample, provided successive runs are conducted in the same manner. Of course, commercial tests should be as accurate as conditions permit when the fulfillment of a contract is in question. It seems to us that the probable errors cited by the author are considerably larger than are justified by experience in reasonably careful commercial testing. They seem to apply to

tests in which some rather obvious precautions are neglected. Unfortunately, there are too many tests of this type which make a pretense of greater accuracy than they possess.

In discussing that part of this paper which deals with probable error we are limiting ourselves to tests in which a high degree of accuracy is sought, such as research investigations, and we base our remarks on experience in conducting several such tests, among which is a test conducted recently, covering a period of several months, the report of which has not yet been made public. In our work we have been at great pains to reduce errors, and yet we believe that the precautions taken are not unreasonably elaborate or costly for any test which pretends to accuracy.

The author states in Par. 3 that water can be weighed with a probable error "well within plus or minus 1 per cent." In our experience we believe that it is possible to attain an error not exceeding 0.1 per cent without unreasonable care or cost, including the use of proper scales frequently checked.

Regarding errors due to a change in temperature of the water in the gage connections, a rough computation indicates that for one of the large Stirling boilers at Connors Creek, a change in the temperature of the gage connections from 400 deg. fahr. to as low as 100 deg. fahr. will lead to an error of only 0.3 per cent of the water evaporated during a run of sufficient length to reduce to 1.0 per cent the starting and stopping errors in estimating the weight of coal in the furnace. This error might be larger for a smaller boiler.

Since our boilers deliver superheated steam, our experience throws little light on the determinations of quality. The author's discussion of the sampling of steam seems to us a very illuminating contribution to this much discussed subject.

With reference to Par. 16, we see no reason why coal cannot be weighed as accurately as water, that is, to within plus or minus 0.1 per cent. A very important factor affecting the magnitude of the starting and stopping errors is the length of run. By suitably prolonging a run these errors can readily be reduced to almost any desired degree. Furthermore, with underfeed-stoker fires, draft readings add greatly in estimating the thickness of the fuel bed. The difference of draft readings over fire and at the damper indicates the rate of gas flow, and if this difference is the same at the start and finish of a run, the rate of gas flow is the same, provided the boiler is equally dirty at both times. Under such conditions the drop across the fuel bed is a fair indication of the thickness of the fuel

bed, provided clinkering does not alter the condition of the fuel bed, and provided that the size of coal is unchanged, a condition which usually prevails in underfeed-stoker practice.

In Par. 17 the author states that allowance should be made for the evaporation of the moisture content of the coal. In this we disagree with him. The necessity for such a correction and the method of making it are by no means well established. The only circumstances under which such a correction is justified in our opinion is in an acceptance test when a previous agreement has been reached as to just how such correction is to be made. In any case the loss of heat from this cause will appear in the heat balance.

With respect to the determination of the moisture content of the coal, much depends upon the method of sampling and handling. Methods have been developed and are in common use which we are satisfied give a result within plus or minus 0.5 per cent.

We have never seriously considered drying all of the coal before firing, and do not see how this could ever be done for a boiler of commercial size, except in installations using pulverized coal.

The author's statement in Par. 18, that much depends upon the method of coal sampling, cannot be overemphasized. We have tried simultaneously three independent methods of sampling, each carefully designed and executed to yield an average sample, and have found the resulting heating values to check well within plus or minus 0.5 per cent. The proximate analyses did not check as closely as this, but these are of less importance. We recognize that if sampling is not done with great care, errors as large as 3 per cent, or even more, may readily be introduced, provided that there is at least that much variation in the character of the coal being burned.

The figures given by the author in Par. 19 we should revise for an accurate research test as follows:

Coal	Error per cent	
Weighing	± 0.10	
Estimating amount of coal in furnace at start and stop	± 0.30	
Total error of weighed coal		± 0.40
Variation between original percentage of moisture and that in laboratory sample + failure of sample to truly represent bulk	± 0.50	
Failure of heating value of sample to represent bulk	± 0.30	
Total error in analysis		± 0.80
Total error in B. t. u. in coal (weight and analysis) on which efficiency is based		± 1.20

WATER		
Weighing, starting and stopping test, gage glass, leaks	± 0.15	
Failure to accurately measure superheated steam temperature, effect on heat absorption	± 0.20	
Total error in measurement of heat absorption		± 0.35

B.t.u. in coal may vary from 98.8 to 101.2 per cent.
Heat absorption may vary from 99.65 to 100.35 per cent.
Reported efficiency may vary from 98.8/100.35 to 101.2/99.65, or 98.5 to 101.6.

The author's allowance of an error of 3 per cent in a carefully conducted test seems to us about twice as large as it should be for a test of research grade with an underfeed stoker.

With reference to Par. 22, we have never thought that a heat balance was used "to confer dignity and precision" to a report, but rather supposed that it was used to do exactly what the author asks that we do, namely, indicate the degree of accuracy attained. We do not believe that any engineer of ability has ever held that the "unaccounted" heat is accurate, but rather that its magnitude is some indication of the accuracy of the test. In a recent test comprising forty runs, the "radiation and unaccounted for" averaged 1.1 per cent. We have computed the radiation to be from 0.8 to 1.0 per cent. Taking this as 0.9 leaves an average error of 0.2 per cent, which indicates that the errors are more or less compensating. In the above tabulation, if we assume that coal and water errors are both in the same direction, we have the reported efficiency varying from 98.8/99.65 to 101.2/100.35, or from 99.14 to 100.84 per cent. Using the author's values, in the same way, we find the efficiency to vary from 98.5 to 101.5 per cent.

The values for variation in the CO_2 content of the gas given in Par. 24 might easily be found in a leaky setting. We have made repeated tests, however, in which the maximum variation was 2.5 per cent. With properly shielded thermocouples, the gas temperature should be measured to within 10 deg. fahr.

Beginning with Par. 25, the author discusses the computation of special "efficiencies" designed to facilitate comparisons of boilers operating under different conditions. We emphatically object to the introduction of such matters in a commercial boiler test. The overall efficiency as now computed expresses fully the accomplishment of the combination of boiler, furnace, and grate, and for strictly commercial purposes no other result is desired. The ratios suggested by the author are details for special investigations in which suitable assumptions must be made in each individual case. In short, our present overall efficiency tells what the outfit does, while the ratios

suggested indicate why it does it. We do not believe that any such segregation coefficients as the author suggests can ever be adopted generally.

A feature of operation which the author does not mention, and which we have found to be of considerable significance, is the burning quality of the coal. We recently conducted tests with two coals substantially identical as regards proximate analysis and heating value, under strictly comparable conditions, with a consistent difference in overall efficiency of one per cent. Another important item is the cleanliness of the boiler surface during the test run.

The questions raised in Par. 33 should logically be covered by the revised Boiler Test Code now in preparation. The boiler-test report should show the amount of boiler surface and superheater surface and the degree of superheat, and therefore makes it possible to make any necessary allowances for differences between boilers. Similar remarks apply to economizer surface.

The author's contention in Par. 35 points to a highly desirable end, but the method of attaining this end is a large question. For the present, at least, a boiler-test report should present complete data concerning auxiliary equipment, such as stoker blowers, stokers, clinker grinders, soot blowers, induced-draft fans, ash- and coal-handling equipment, etc., external to the boiler and furnace. Then anyone wishing to compute a net efficiency can do so on any basis which meets his needs or his fancy. In our opinion, a boiler-test report should always show the gross overall efficiency, and, if net values are given, the method of computing them should be made very clear.

The tabulation in Par. 38 is a remnant of historical practice. Most people think in these terms, and this is ample justification for retaining them for the present. We grant that some, if not all, of them have small real significance.

H. Kreisinger. The errors in making boiler tests are frequently as large as the author gives them, and in some cases are even larger. For that reason, the results of a single boiler trial ten or even twenty-four hours long should be accepted with reservation. Only when we make several trials of twenty-four hours' duration and at the same rating can we be reasonably sure of the results as given by the average. When it is desirable to know the performance of a steam generating unit, several groups of tests should be made, each group consisting of two or more boiler tests at the same rating, and the different groups at different ratings.

If the results are plotted on cross-section paper, with the rating as an abscissa, and a smooth curve is drawn through the points, such a curve can usually be depended upon to give the true results to one per cent.

The weight of water evaporated on a boiler test of twenty-four hours' duration can be determined usually with a fair degree of accuracy. Most of our modern steam-generating units contain superheaters, and therefore the quality of the steam can be determined by measuring the temperature of the superheated steam. Therefore the question of sampling steam, and the kind and location of the steam calorimeter is losing its importance in boiler tests. However, the author's reasoning about the steam sampling tube for determining the quality of saturated steam sounds logical, and those who are interested in that particular feature should look into it.

In the determination of the temperature of superheated steam, we are not free from possible errors. On large steam-generating units, there are two outlets from the superheater and the temperature must be measured in both outlets. It frequently happens that one of these outlets shows a temperature of 20 to 30 deg. fahr. higher than the other. Obtaining the true average temperature would be a simple matter if we were sure that the same weight of steam flowed through each outlet. But we are never sure of that. Steam flow meters placed in the two outlets frequently indicate 10 to 15 per cent greater flow from one outlet than from the other. It is not always the outlet with the lower flow that has the higher temperature. Sometimes the temperatures in the two outlets reverse several times during a run of twenty-four hours. However, it does not appear that the error from these different temperatures would be large. Assuming that one outlet discharges 15 per cent more steam than the other, and that the temperature is 30 deg. fahr. lower; that 15 per cent of the steam would be 15 deg. fahr. below the average temperature, which is an error of only about 0.1 per cent of the total heat absorbed by the boiler and the superheater.

The weight of coal burned is comparatively easy to determine to within less than 0.5 per cent, particularly on a twenty-four-hour test.

The largest error entering into a boiler test, in the writer's opinion, is in the collection and the preparation of a coal sample. This is particularly true of stoker coal. A representative sample of powdered coal is comparatively easy to collect. A coal sample collected in the boiler room weighs 200 to 500 pounds. The sample on which the chemist makes the ash and B.t.u. determination weighs

one gram, which is 1/100,000 to 1/250,000 part of the sample collected in the boiler room. The greatest error occurs in the process of getting the one gram out of the boiler-room sample. Errors of 2.5 per cent may easily occur. Even after the boiler-room sample has been reduced to ten pounds and the sample is divided in two, and the two parts are sent to different laboratories, the reports from the two laboratories may be 2.5 per cent apart. One does not need to examine very many reports of chemical laboratories to find errors of this magnitude. Table 1 contains the results of ash and B.t.u. determinations on three samples, duplicates of which were

TABLE 1

Sample	No. 1		No. 2		No. 3	
Laboratory	A	B	A	B	A	B
Ash in dry coal	19.03	16.67	19.60	18.89	18.82	15.19
B.t.u. in dry coal	11363	11740	11279	11351	11401	11945
B.t.u. in moisture- and ash-free coal	14033	14080	14029	14000	14045	14070

sent to two different laboratories. The smallest difference in the percentage of ash as determined by the two laboratories is 0.71 per cent and the largest difference is 2.63 per cent. Attention is called to the fact that the heat values based on the moisture- and ash-free coal agree very closely for both laboratories. It can be safely assumed, therefore, that the men making the calorimetric determinations were doing their work accurately. But the two grams of coal the calorimeter men used in their determinations contained different percentages of ash and therefore gave them very different calorific values for dry coal. The error occurred in the interval from the time the boiler sample was reduced to ten pounds to the time a one-gram sample was taken from each five-pound sample, which two samples were supposed to have been duplicates. It might have been done by the man who divided the ten-pound sample into two parts, or by the men in the two chemical laboratories who reduced the five-pound samples, or by the two calorimeter men when they were taking their gram of coal after the samples had been prepared in the laboratories. The three samples were collected in the same

boiler tests, and presumably the same coal. The ash determinations made in laboratory A agree within less than one per cent. Apparently the preparation of the coal in the boiler room was fairly careful. It was apparently fairly carefully done in laboratory A. The indications are that the error was made in the preparation of the coal sample in laboratory B.

Boiler efficiencies calculated on the basis of the reports from the two laboratories may differ three per cent. Thus, if the heat absorbed per pound of dry coal was 8500 B.t.u., the efficiency, figured with sample 3 and laboratory A determinations, would be 8500/11401, or 74.5 per cent. With the B.t.u. determination from laboratory B, the efficiency would be 8500/11945, or 71.1 per cent.

Table 2 shows a discrepancy between two ash and B.t.u determinations made in laboratory A on presumably the same sample of coal. The second sample marked "check" was part of the "original" sample held in reserve. It was held several days and then sent to the laboratory. The chemist did not know that it was a check sample on the "original." If the boiler absorbed 8500 B.t.u. per pound of dry coal, the efficiencies, calculated on the two B.t.u. determinations, would be as follows:

TABLE 2

Sample	Original	Check
Ash in dry coal	17.30	18.04
B.t.u. in dry coal	11579	11391
B.t.u. in moisture- and ash-free coal	14001	14001

Original, 8500/11579, or 73.4 per cent
Check, 8500/11391, or 74.6 per cent

It should be added that the above discrepancies in the ash and B.t.u. determinations were picked up at random in connection with some recently made boiler tests with stoker-fired furnaces. If one should look particularly for discrepancies in coal analyses, he would find some worse than those given in Tables 1 and 2.

The best way to reduce the effect of the errors in sampling and analyzing coal is to take a number of samples on each test, and

prepare and analyze them separately. The average of the analyses should be used in calculating the results of the tests. At least four, and preferably six, samples of coal should be taken on each test. Moisture and ash should be determined on each sample. B.t.u. determinations, and proximate and ultimate analyses can be made on a composite of all the samples. The B.t.u. value should be reduced to the average of the moisture and ash determinations of the six samples. On important tests, and all tests are important if they are worth the time required to make them, one should never depend on one or two coal samples. The cost of collecting, preparing and analyzing coal samples is small when compared with the total cost of a well-conducted boiler test. It pays to put a well-trained man who knows what he is doing on the task of looking after the coal sampling, and he should have nothing else to look after.

The heat balance on a boiler test, if sufficient data have been collected to compute it with a fair degree of accuracy, is perhaps the most valuable part of a report of a boiler test. It shows what the principal losses were and indicates the accuracy with which the test was made. After accounting for the heat absorbed by the boiler, and all the losses, including the radiation, there is usually a quantity left to make the heat balance add to 100 per cent. This quantity may be either plus or minus, and can be called "errors and unaccounted for." It contains all the errors made on the boiler test. This quantity will be zero if no errors have been made on the test. It will also be zero if the algebraic sum of all the errors is equal to zero. It therefore does not mean much if the "errors and unaccounted for" sum up to zero on one test. There may be large errors with opposite sign which will tend to neutralize each other. Thus, for example, the efficiency may be five per cent too high, the loss in the dry flue gases and the loss in the combustible in the ash each 2.5 per cent too low. These errors will sum up to zero, and make the item "errors and unaccounted for" appear zero, or very small. If, however, on a series of tests this item remains small, though it may be on both sides of the zero line, it is a fairly good indication that the tests were made and calculated with care, and that the errors are small.

The writer agrees with Mr. Cotton in the matter of reporting the results of boiler tests directly in B.t.u. absorbed per pound of coal, or B.t.u. absorbed per 100 B.t.u. in the coal. The "equivalent evaporation" and the "factor of evaporation" are clumsy to handle and are meaningless in our modern power-plant practice. They

should be dropped as a useless burden. Furthermore, why use a unit that changes with time? In the days of Peabody's Steam Tables, one pound of equivalent evaporation meant 966 B.t.u. Today it means 970.4 B.t.u., and the indications are that before long it will mean 972 B.t.u. Why not adhere to the B.t.u.? If we wish larger units, why not use 1000 B.t.u. as a unit? Surely 8500 B.t.u. means more to most of us than 8.76 pounds of equivalent evaporation.

The rate of working can be conveniently expressed in B.t.u. per square foot of heating surface per hour. This method would enable us to have uniform expressions for the rate of working the boiler, superheater and economizer.

R. F. Burke. The writer is of the opinion that a better steam sampling nozzle could be designed and that there are entirely too many holes in the nozzle recommended by the Society's test code which contains twenty 1/8-in. holes. He suggests a calorimeter nozzle having three openings, one in the center and one near each side. Calorimeter nozzles should be proportioned so the velocity of the steam would be the same through the calorimeter nozzle and calorimeter and the total amount of steam entering the calorimeter nozzle should be passed through the calorimeter.

With large stoker-fired boilers when the furnace is perhaps 15 to 20 ft. or more wide, it is almost impossible to judge the amount of coal on the grates at beginning or end of test. Usually the amount of coal on the grates of underfeed stokers is judged on the depth of the coal at the observation doors. It is not at all difficult to have much more coal in the center of the fire, stored well up in front and not noticeable to the observer judging the thickness of the fire. With chain-grate stokers this cannot occur. The writer is of the opinion that a twelve-hour test with a chain-grate stoker is more accurate than a twenty-four-hour test with an underfeed stoker, and he believes the correct way to judge the amount of coal on the grates at the beginning or end of a test with an underfeed stoker, is by the draft differential.

Determining the water fed to a boiler can be done fairly accurately; the fact is, there are numerous meters on the market which have an error of less than $\frac{1}{2}$ of 1 per cent but there is an error which can occur in the amount of water fed to boilers which has nothing at all to do with scales or meters. If the water was high for, say, 15 minutes before the beginning of test and a large amount of water had to be pumped into the boiler to bring the water

up to its proper level at the end of test, a large error could take place. To overcome this the water should be carried at a constant level for 15 minutes before the test and then carried at that level throughout the test if possible. Pumping a large amount of water into the boiler at the end of the test should not be permitted.

Usually if a test figures up showing a proper heat balance it is considered accurate, whereas, there can be a large error, even up to 5 per cent. One of the great difficulties in obtaining proper information to make a reliable heat balance is to obtain the amount of combustible in the ash. With a large boiler running a twenty-four-hour test burning, say, 200,000 lb. of coal with possibly 20,000 to 25,000 lb. of ash, it is almost impossible to obtain a representative sample, even if the entire amount of ash is broken up and the quartering method recommended in the Society's code is used, yet it is one of the most important items in connection with the test, for a large amount of this ash may be combustible matter. It is not fair to the boiler if the analysis of the ash is omitted because the efficiency of the test would then depend more upon the stoker and the operator than upon the boiler.

The writer is of the opinion that the efficiency of boilers should be determined more upon temperatures and flue gas analyses than on the amount of coal burned and water evaporated. It is quite true that the composition of the flue gas varies across the width of a boiler. This, of course, depends a good deal upon the tightness of the setting and sometimes on the design of boiler and baffling arrangement. Usually the gas is taken from the center of the flow of the gases in the center of a boiler. This undoubtedly gives the highest CO_2 and might be recorded as the maximum CO_2 in the flue gas.

The writer is of the opinion that the superheating surface should be added to the boiler heating surface and the rating of the boiler obtained by allowing 10 sq. ft. per boiler horsepower for the total. As for economizers, when they are an integral part of the boiler, economizer heating surface should be totaled and the rating obtained by allowing 10 sq. ft. per boiler horsepower for the total. The writer does not see how the back bank of tubes in a Stirling, Badenhausen or any other type of boiler can be considered separate from the boiler.

W. D. Ennis. Mr. Cotton's estimate of a plus or minus 3 per cent error in a "regular carefully conducted boiler test" is more or

less justifiable according to what is meant by a carefully conducted test and particularly according to the length of such test. No doubt there are many boiler trials made, the accuracy of which is not closer than plus or minus 5 per cent. In fact a boiler test made with coal fuel never is and never can be accurate. Besides all the other reasons for this the difficulties of sampling and analyzing the fuel are sufficient.

While the author does not make specific recommendations toward the increase of accuracy it is quite apparent from the paper, although not brought out very strongly in the discussion, that the primary requirement is a test of reasonably long duration. No duration however long will eliminate all of the errors but every added hour of the run, assuming uniform care and attention, reduces the proportion of total error. The one big error in a boiler trial is made when it is too short. A forty-eight-hour run is in several respects more accurate than the average of two twenty-four-hour runs.

The so-called heat balance is too useful to give up, but perhaps it would be wise to state only the computed items and their total as representing the "heat accounted for."

There are altogether too many boiler tests made and a large proportion of them are perfectly useless for many purposes. As pointed out by Mr. Burke, studies of temperature and flue-gas composition often give more reliable information than the evaporation. The writer recalls the instance of a coal-feeding device, the only function of which was to spread buckwheat coal over the grate. The merit of the device could have been determined by an experiment made 100 miles from any boiler and the actual tests made by using the machine under a boiler indicated several interesting things but only incidentally the functioning of the machine. It is extremely illogical and unwise to use a steam boiler as a measuring instrument. It is costly, inaccurate and subject to all sorts of disturbing influences, to say nothing of the human factor which always enters into its operation.

Daniel Adamson. P. W. Williams in 1893 read a paper before the Institution of Civil Engineers (Eng.) which is recorded in their *Proceedings* CXIV, p. 28, in which he showed leakage to be 1.9 per cent.

In the *Proceedings* of the American Institute of Electrical Engineers, December, 1910, p. 1681, is a paper by Dickinson and

Robinson, who mention tests where the boiler leakage amounted to 10 to 12 and up to 20 per cent.

The British Admiralty tests referred to by the author are probably those carried out in 1900 and published in *Engineering* (London) Aug. 5, 1904, p. 192.

The Author desires to express his gratification that the main object of the paper has been so amply fulfilled by the masterly discussion of leading authorities. This discussion has been so thorough that it is impossible to reply to it fully without writing another paper.

Replying to Mr. Bell, the author prefers to have all test reports accompanied by heat balances. They indicate the value of the reports. Serious unbalance shows inefficiency of observation or of operation. Reports of a series of tests wherein the "unaccounted for" was consistently over 20 per cent were recently submitted to the author, who had reason to believe that much more coal was leaving the fire in a non-gaseous state than was proper and than the ash analysis suggested.

The value of the heat balance as a statement of what happened to the heat is really not open to question. But many engineers who are not skilled in boiler testing, are unduly impressed with the heat balance, and the author, therefore, feels justified in repeating that it is not a *balance* and that it adds nothing to the accuracy of the test. It is usually less of a check on the efficiency than on the other items.

The author appreciates Mr. Bradshaw's remarks on the determination of dust in blast furnace gas, and its similarity to the determination of water dust in steam. That the velocity of the gas sample entering the tube should be the same as that of the bulk agrees with the author's views; as also the use of a sampling tube pointing upstream.

Mr. Burke suggests using the draft loss through the fuel bed to determine the condition of the fire at start and stop. This is a step in the right direction; but the presence of clinker must be reckoned with, and variation in the size of coal in some cases, as these would increase the draft loss for the same amount of fuel in the bed.

His caution about the rate of feed at start and stop is very important. It is probable that many engineers would be surprised at the error which can result in this way. It is more accurate to

calculate the deficiency at the end of the run than to pump rapidly up to the mark as the test is coming to an end.

The importance of having conditions as well as quantities the same at both ends of the trial cannot be stressed too much.

The difficulty of getting even an approximate idea of the proportion of fuel sent to the ashpit may be the cause of considerable error in the heat balance as has also been pointed out by Mr. Bell.

It is questionable whether it would be agreeable to rate all superheaters at 10 sq. ft. to the hp. Should Mr. Bell's radiant heat absorption superheater be rated at this figure?

Replying to Mr. Funk, the author's reasoning on the reliability of steam sampling is based very largely on experience with other apparatus. Experimental investigation is highly desirable.

Messrs. Hirshfeld, Berry, Thompson, and Carter have done a valuable work in pointing out that many lawsuits have followed this false conception of accuracy. These would be avoided by our admitting that we cannot guarantee the accuracy of boiler trials to be within some given percentage, and then setting definite tolerances to be allowed. This naturally refers to commercial tests. Research tests have little to do with guarantees, and nothing to do with lawsuits.

Research tests are made at home, so to speak, and under conditions which cannot usually be duplicated profitably with commercial tests. The author believes that when these gentlemen, who have devised and frequently operate highly developed testing methods and means in the same plants, place the accuracy of these precision tests at about plus or minus 1.5 per cent, they virtually confirm the author's proposed allowance of plus or minus 3 per cent on a commercial test. He does not believe it is generally recognized that such research tests may be out 1 per cent.

There is another kind of research test, the kind made by the ambitious engineer of the small plant, without much apparatus or abstruse knowledge. The author recently saw a case where such an engineer had raised the "actual" evaporation per pound of coal "as fired" about 15 per cent as averaged from a large number of tests. These tests were scarcely worth the name and probably carried large errors, but the fact remains that his employer is buying less coal than he used to.

The use of the boiler as a gas flow meter in connection with the pressure drop through the fuel bed is excellent in checking the thickness of the fuel bed, and may reduce this error very materially.

The author did not in his Par. 17 refer to drying coal *to be fired*, but to drying samples to ascertain the moisture while the test was in progress.

The author entirely agrees with Mr. Kreisinger that the really important thing is not a single test, but a load-efficiency curve, drawn through a group or groups of tests. The accuracy of such a curve may easily be within 1 per cent. But such multiple testing is not always possible with commercial trials.

To emphasize the care necessary in taking samples it may be well to mention that the coal analyzed in the laboratory is of the order of 1/10,000,000 of the coal burned and that about 1/100,000,000 of the flue gases are analyzed. The serious feature of errors in the heat value of the fuel is that they are in no wise reduced by increasing the duration of the test. Unless the heat value error can be made very small, a point is soon reached where little is gained by increasing the duration of the trial. Mr. Kreisinger's broad discussion on the reliability of the heat value is rendered the more authoritative by his long experience.

The author feels prompted to add the following recent analyses of fuel oils:

Analyst	B.t.u. per pound		
	Boiler Test Number		
	4	7	8
A	.....	.....	17,803
B	.....	.....	18,600
C	17,081	17,261	17,199
D	.....	17,295	.....
E	.....	16,980	.....
F	.....	.....	.....
G	17,334	16,968	17,256

In test No. 4, analyst G is 1.48 per cent above analyst C, while in No. 7, C is 1.72 per cent above G. In No. 8, they agree more closely but analyst B gets a result 2.35 per cent above G.

The "boiler horsepower" dies slowly. If we were to express the load in all future test reports as B.t.u. per sq. ft. of heating surface per hour as Mr. Kreisinger suggests as well as in "percentage of rating," it is probable that we could become reconciled to the loss of the boiler horsepower more quickly.

Mr. Davison mentions handhole leakage as one of the unknowables. The author believes that during the trials made about twenty-five years ago by the British Admiralty to determine the relative merits of Belleville, Scotch and other boilers, the water loss in the Belleville ran about 5 per cent of the evaporation in some instances, but this is only from memory.

The author does not agree with Mr. Davison that the manufacturer and customer should settle upon a tolerance. The first manufacturer who tries to do this is very liable to be looked upon as having little faith in his own product. Such tolerances must be settled and recommended by the Boiler Test Code Committee if at all, and it is to be hoped that this will be done.

The author is glad to find Mr. Tenney in agreement with him that all conditions surrounding the test should always be reported. This is not done very often, and some important conditions are scarcely ever reported. It is obvious that a certain efficiency attained with the coal Mr. Tenney burns might be much more creditable than a little higher efficiency with a much better coal.

In reply to Mr. Vennum, the present boiler efficiency shows what proportion of the heat in the fuel was used to make steam, but it is reasonable that an efficiency should also be reported which shows how much heat was used of that which it was possible to use. This proposed efficiency would be mainly for research purposes, not for commercial tests.

Mr. Vennum says that we want to know what proportion of the heat in the fuel is made available for mechanical use, and it is obvious that this is just what our present efficiency does *not* show us, until after we have deducted the auxiliaries.

Professor Ennis very wisely calls attention to a very common absurdity, — that of using the boiler as an instrument to measure the mechanical perfection of a coal firing machine. The good or bad features of the device as a machine will not be disclosed by a regular boiler test, which may even impede the desired observations.

The author agrees with Professor Ennis that a large proportion of boiler tests are perfectly useless for many purposes. As Mr. Tenney pointed out, the whole of the circumstances surrounding a test should be reported. Those who have collected a number of boiler test reports with a view to investigating some particular feature will remember finding that perhaps only 5 per cent of them contained enough data to allow of reasonably safe comparison. Without some description of the apparatus used and the care taken

to avoid serious error, the boiler test report carries no evidence on its face as to its accuracy; and the investigator automatically looks for and relies on the name of the person who conducted the trial. The heat balance is not a balance and does not add to the precision attained. Its unbalance generally throws less discredit on the efficiency than on the rest of the items. There is no question of giving up the heat balance — that is quite impossible because it simply happens naturally. But Professor Ennis's suggestion is certainly valuable. Make some allowance for radiation and add the thing up without any "unaccounted for," and its difference from 100 will stand out clearly and tell us what we want to know.

Speaking generally, the discussion confirms the author's opinion that the inaccuracies at present inherent in commercial boiler testing are sufficient to warrant the introduction of tolerances in the performance of guarantees as a recognized engineering custom. In research testing, the errors can be reduced to about one-half of those of commercial tests, and to still less than this as the number of the tests is increased and rendered into a load-efficiency curve.

No. 1842

USING EXHAUST ENERGY IN RECIPROCATING ENGINES

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and
C. C. Trump, Syracuse, N. Y.
Members of the Society

There are a surprising number of cases where oil, gas or steam engines will show better economy than purchased power and electric motors, and this may be emphasized where the exhaust from steam engines is valuable for heating or process work. Any development which will improve the efficiency of reciprocating engines, and at the same time simplify their construction should be of interest.

Up to now, reciprocating engines have suffered from losses due to incomplete expansion. There has been a compromise between such losses and mechanical ones connected with the use of very large cylinders. Oil engines, multiple-expansion and uniflow steam engines are coming to use high ratios of expansion, and these are doing much to reduce exhaust losses. Many advantages may be gained, however, if the energy remaining in the exhaust at release can be converted into kinetic energy, and from that into partial vacuum in the cylinder. In double-acting uniflow steam engines, for instance, pistons and cylinders may be made smaller and shorter for a given load, the number of exhaust ports reduced, and a greater range of back pressures may be negotiated without extra clearance or exhaust valves. By attention to design of exhaust pipes and nozzles the performance and efficiency of such engines may be a good deal improved.

The theoretical problems are presented in this paper for discussion, and practical applications to either single-cylinder or multi-cylinder engines are suggested and illustrated.

WITH the rising prices for fuels in this country, anything which may result in fuel economy through turning more heat into useful work should be of interest.

2 The use of kinetic energy in long exhaust pipes has often been attempted in a practical way to the writers' knowledge in gas, gasoline and oil engines, especially of the two-stroke-cycle

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type, to improve the scavenging effect and increase the power output. Successful results were usually obtained more from a method of trial and error than from any attempt at design, or the use of nozzles, diffusers, etc., which might avoid losses and improve the effect.

3 Similar long pipes have been used on the suction side of air compressors, and on the discharge of pumps, to improve the volumetric efficiency. Whether conscious attempts have been made to produce such effects in ordinary steam engines, which usually have exhaust valve passages too tortuous to avoid serious losses, will no doubt be brought out in discussion. It is probable that the studied application of suitable exhaust pipes with properly designed nozzles for gas and steam engines, especially those with

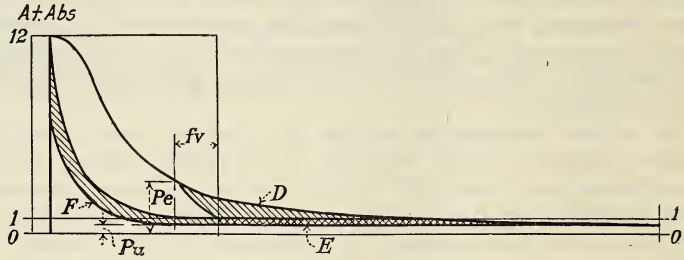


Fig. 1 Conventional Steam-Engine Indicator Card Showing Exhaust Energy

piston-controlled exhaust ports, is new, for a United States patent has just been allowed on it.

4 In Fig. 1 is shown a conventional indicator card from a steam engine, with the loss due to incomplete expansion represented by the shaded area, *D*. It will be noted that this area is large as compared with the small part of it due to opening the exhaust ports, or valves, considerably in advance of the end of the stroke. If the energy represented by *D*, plus that due to expansion to a lower final pressure such as the area *E*, can be converted into partial vacuum, it will obviously be transferred, or at least all of it but the losses, to the compression side of the card as useful work, represented by the area *F*. This would have the effect, evidently, of reducing the back pressure in the cylinder of the engine and, therefore, the final compression pressure, by a greater amount, the more energy there is available at release.

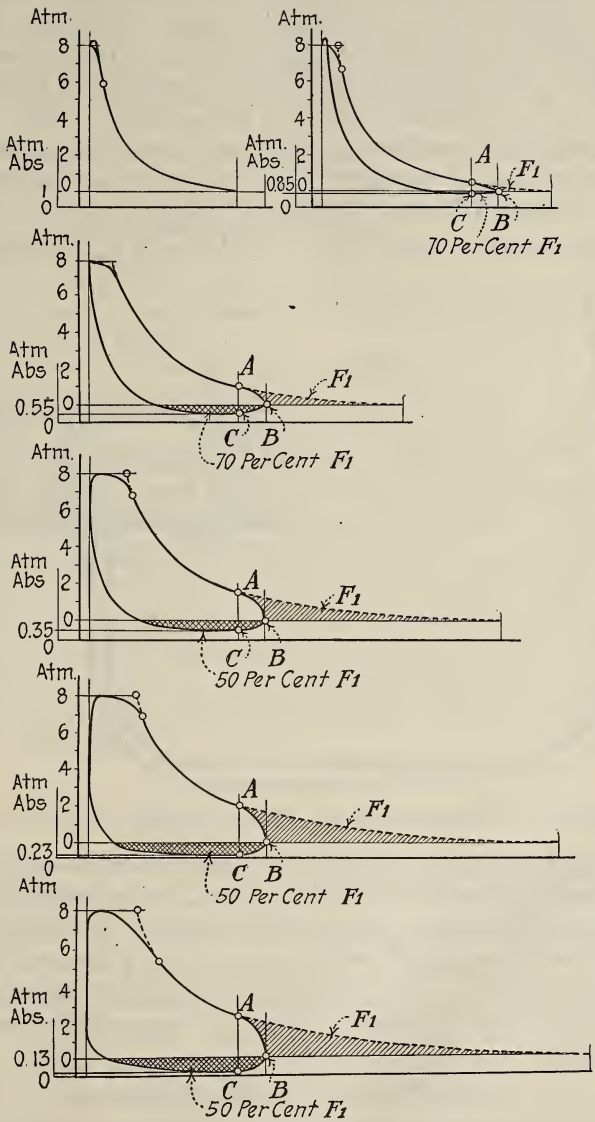


Fig. 2 Cut-Off and Release Pressure Effects on Incomplete Expansion Loss

5 The several diagrams in Fig. 2 show how the loss due to incomplete expansion would increase with longer cut-off and higher release pressures. They also indicate how these losses may be turned into gains in useful work by reduction of back pressure and compression. The cards are all drawn for exhaust to atmospheric pressure and the pressure scale is in atmospheres. They indicate that any method of transferring energy into partial vacuum must be able to take care of variable loads. They do not indicate any relations between the time of exhaust opening and the velocities used in the exhaust pipes. These would have to be shown in a

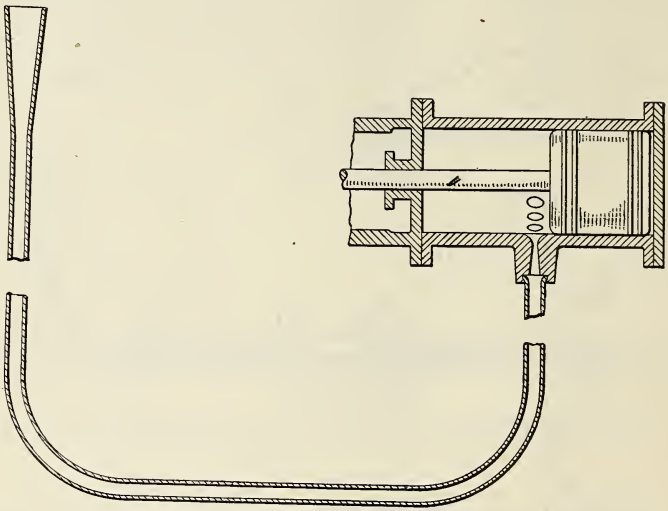


Fig. 3 Exhaust Nozzles and Long Pipe of Uniflow Steam Cylinder

different kind of diagram, based on energy rather than pressures. The problem is somewhat similar to those met in ballistics, or in the design of Humphrey gas-pumping engines, in which kinetic energies of solid and elastic bodies have to be figured with respect to time and distance. The number of variables is so many, that graphic rather than numerical methods of computation are to be recommended.

LONG EXHAUST PIPE IN PRACTICE

6 In order to give an idea of the practical application of a long exhaust pipe, for the above purpose, a uniflow steam cylinder with a number of exhaust nozzles and a long pipe is shown in Fig. 3. The nozzle in the cylinder is difficult to design for best

efficiency because it is opened relatively slowly by the piston near the end of its stroke. The other end of the pipe should have some kind of diffuser for reducing velocity-energy to pressure-energy, and this is also not easy to design. The action of the pipe itself would seem to depend upon the following several variable factors:

a The speed of the engine and number of exhaust puffs per minute.
b The variation of pressure at and after release at the cylinder.
c The variation of pressure at the other end of the pipe.
d The area of the nozzles and exhaust pipe.
e The length of the pipe.
f The duration of the exhaust period till closing of ports.
g The compression and expansion of the steam or gas itself.
h Heating due to friction.
i Cooling due to outside temperature conditions.

7 It may have been the experience of many, as it has been of the writers, that a pipe of this kind, subject to sudden puffs, will cause the steam or gas in it to pulsate. It is sometimes evident in cards taken with sensitive indicator springs. Its length and volume relations may be likened to those of an organ pipe. The difference may be pointed out that the pressure changes and velocities may be higher in this case, and vibrations much slower and longer are desirable than would be the case in the production of sound waves, which are to be avoided with exhaust pipes. It is evident, however, that the slower the speed of the engine the longer must be the pipe.

PROVISIONS FOR LOAD VARIATION

8 For engines running at moderate and nearly constant speed, the pipe length may be fixed, and the area designed to give the best suction effect at the most desirable load. Variation of load can be provided for, as shown in Fig. 4, with extra nozzles and exhaust pipes which may be thrown in or out of action by the engine-governing system. Another method would be to provide an exhaust pipe in the form of a coil, with valves which would change connection between cylinder nozzles and diffuser so as to vary the length of the pipe. This might be necessary with variable speed engines.

9 With single-cylinder engines the length of exhaust pipe necessary to produce the maximum suction effect may be too great for practical uses. As with a musical instrument, however, each

vibration of the medium in the pipe may have its harmonic; or at least will be followed by other pulsations of reduced amplitude. If the second or third pulsation is caught at its lowest pressure as the exhaust ports close, instead of the first, the pipe will have to be only one half, or one third as long, respectively. The effect of friction losses and cooling may change the exact length, but the result may be in favor of the shorter pipe. Fig. 5 illustrates the kind of a vibration which may be expected.

10 Calculations for the proper length of exhaust pipes involve

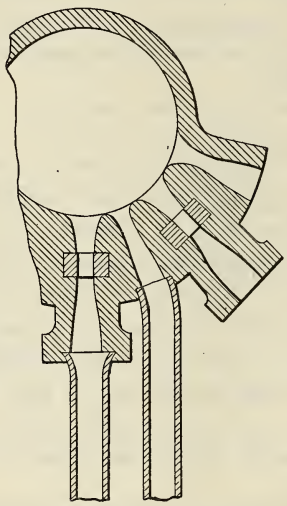


Fig. 4 Extra Nozzles and Exhaust Pipes for Regulation under Load Variation

so many variables that it is probably easier to choose a reasonable size pipe and arrange the cylinder with approximately the proper nozzles, then cut and try with different lengths of pipe and with a sensitive spring indicator on the engine until the best length is found. It may be interesting to suggest the use of graphical curves of pressure-volume, energy-volume, and energy-time as a means of arriving at the approximate design with which to start experiments.

OVERLAPPING EXHAUSTS

11 For multiple-cylinder engines, especially those which run at high speed, the periods of exhaust will overlap, and by proper design of nozzles, one exhaust may be made to draw on the other.

Such a twin-nozzle design is indicated in Fig. 6, which is that on which the work with a two-cylinder locomotive has been based. The overlapping of the exhausts for such a two-cylinder double-

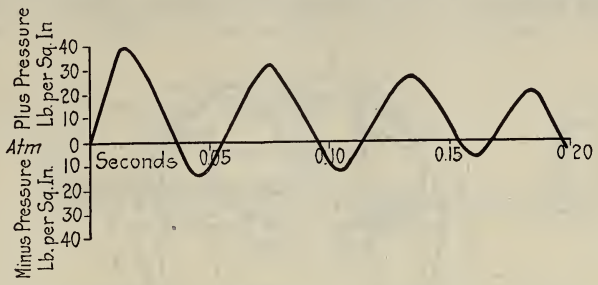


Fig. 5 Probable Vibration of Medium in Exhaust Pipe of Single-Cylinder Engine

acting engine is shown in Fig. 7. The gain in overlapping with a three-cylinder, double-acting engine is shown in Fig. 8.

12 It is evident from these two diagrams that with four ex-

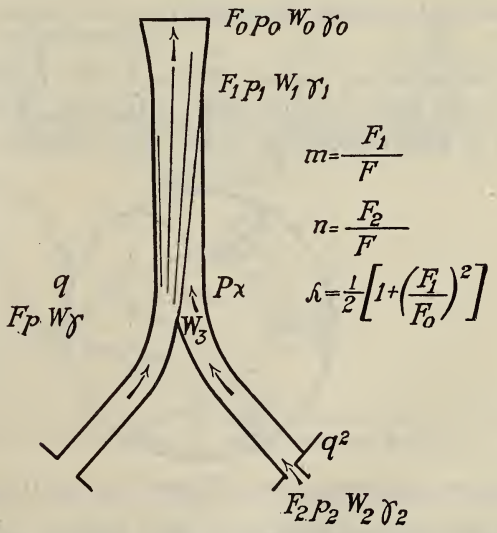
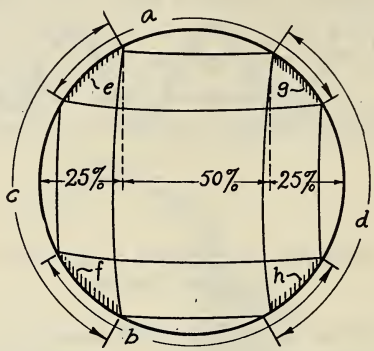


Fig. 6 Twin-Nozzle Design for High-Speed Multiple-Cylinder Engine

haust puffs per revolution, or less, it is necessary to take account of the vibratory action of the exhaust pipe. Its length and volume and the cooling and friction loss must be fixed either by calculation or by experiment, or in combination. With six or more exhaust puffs per revolution the suction effect may be almost entirely

produced by ejector nozzles, so paired as to coöperate and induce partial vacuum in each cylinder at the instant when the exhaust port is closed, at the beginning of compression. Even in this case, it may be necessary to proportion the exhaust pipes, in length and



Fig, 7 Overlapping of Exhausts for Two-Cylinder Double-Acting Engine

volume, so that adverse vibrations may not be set up to spoil the vacuum.

13 Single-acting engines of four cylinders and less will evidently require longer exhaust pipes than those of six cylinders and more. The higher speeds at which these engines are usually run

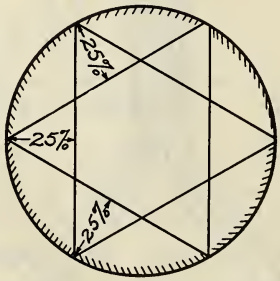


Fig. 8 Gain in Overlapping With Three-Cylinder Double-Acting Engine

will tend to reduce the necessary pipe length. All the benefits of the exhaust ejector action accrue to single-acting engines, except that when the pistons control the exhaust ports they cannot be made shorter, as will be explained later, with respect to this advantage, in double-acting uniflow engines.

14 Ordinary counter-flow steam engines with exhaust valves, or with a single valve for admission and exhaust, not only would

be difficult to arrange with suitable nozzles to avoid eddy losses, but the exhaust period is usually made too long to be covered by the larger pulsations of the long pipe, or the ejector effect of other nozzles. A cylinder with exhaust ports, like a two-stroke-cycle gas engine, or a uniflow steam engine, can be provided with almost

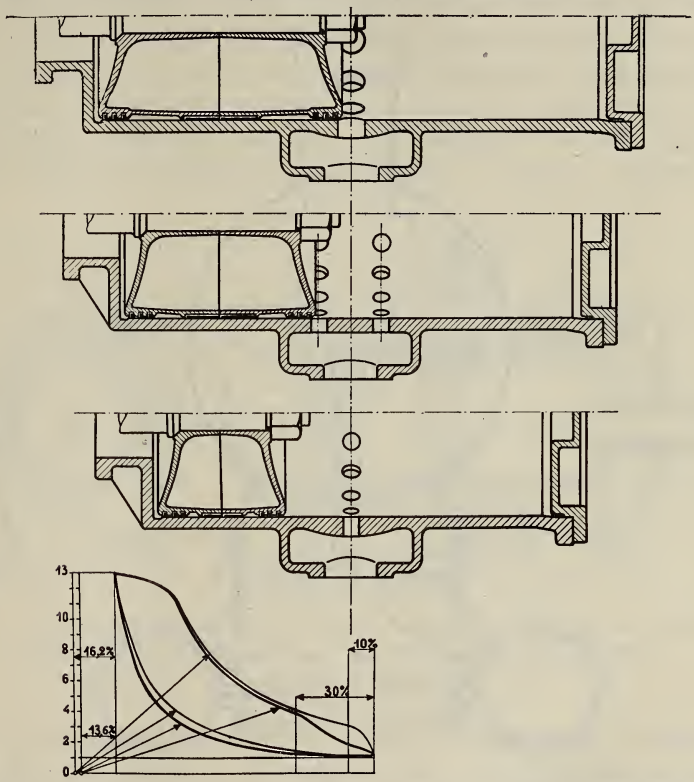


Fig. 9 Area Modification and Location Rearrangement to Shorten Piston and Cylinder of Double-Acting Engine

unlimited exhaust port area, and the exhaust period can be controlled, to be long for high speeds, or short for low speeds. The four diagrams in Fig. 9 show how the exhaust ports may be modified as to area, and rearranged as to location, so that, with a double-acting engine, both the piston and cylinder become shorter. For non-condensing work, such as in locomotives, automotive vehicles, and a large proportion of our stationary engines, the number of ports may be reduced to one, two or three, and the exhaust belt of

large diameter dispensed with as shown in Fig. 10. The smaller nozzles in this illustration are provided for bleeding steam, after it has done useful work, for heating cars, or feedwater. Such nozzles may also be used for supplying steam for process work, or for heating, with a small part of the exhaust from stationary engines. The long exhaust pipe and ejector action may also be

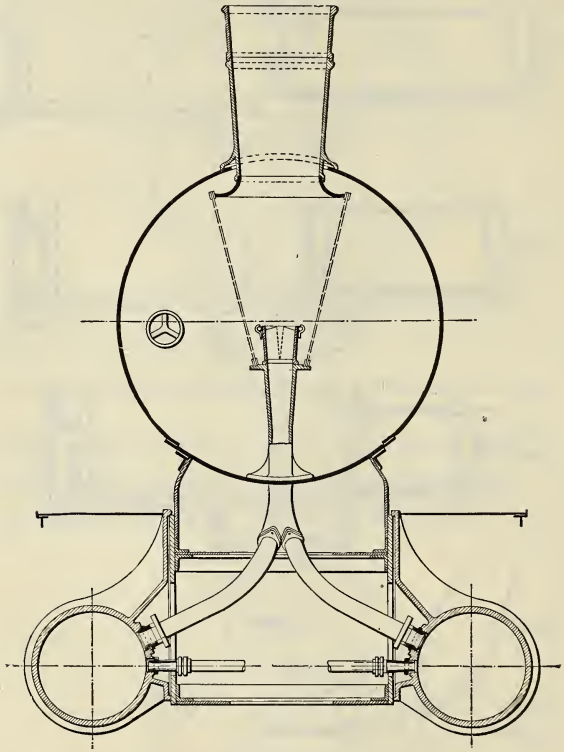


Fig. 10 Number of Exhaust Ports Reduced and Large Diameter Exhaust Belt Dispensed With in Non-Condensing Engine

used to reduce the clearance volume required to control compression, or to dispense with exhaust valves, when it is desired to operate the engine with exhaust against back pressure higher than atmospheric, and use all the steam for heating or process work.

COMPARISON OF CYLINDER DESIGNS

15 The practical advantages of reducing the exhaust orifices to two long slots are well illustrated by comparison between Figs.

11 and 12. Fig. 11 shows the original design of cylinder, for a portable engine and boiler, for farm or well-driller use. The long piston, large number of exhaust ports, large exhaust belt, and the necessity for compression-relief valves, are evident. In Fig. 12, however, the piston and cylinder have been shortened considerably, the piston over 25 per cent, and the cylinder nearly 20 per cent. The exhaust ports and belt have been reduced to two nozzles, and the cylinder support has been simplified and moved to one end only. Compression-relief valves have been dispensed with, except for small drain cocks for starting a cold engine. The long exhaust pipes are shown leading to the smokepipe, in Fig. 13. Experience

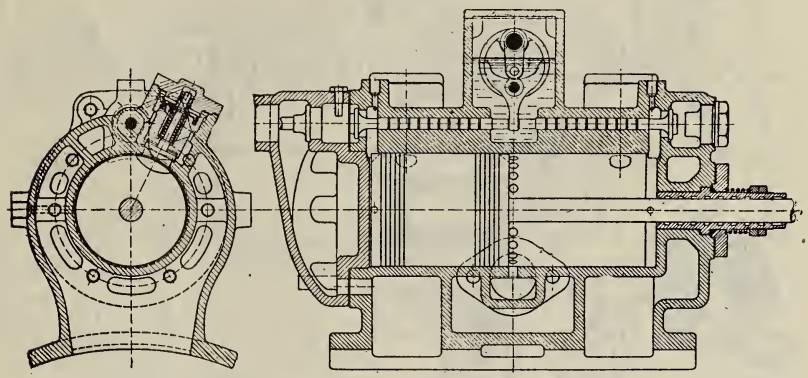


Fig. 11 Original Cylinder Design with Long Piston, Numerous Exhaust Ports, Large Exhaust Belt and Necessary Compression-Relief Valves

with the locomotive has shown a more uniform draft is induced than was the case with the sudden puffs of the ordinary exhaust-pipe arrangement. The pipe shown leading out of the diffuser in the smokestack is evidently provided to take off steam for heating feedwater.

NOZZLE ARRANGEMENT

16 In multi-cylinder engines several different arrangements of exhaust pipes and nozzles suggest themselves. Perhaps the simplest is that often seen in gasoline-automobile-engine exhaust pipes, which may be called a series arrangement as shown in Fig. 14. This may have the disadvantage that the cylinders farthest apart should benefit by overlapping periods of exhaust, whereas the several nozzles between the end cylinders are likely to

interfere with friction losses. Bringing a loop of exhaust pipe back behind the first nozzle, as shown dotted in the diagram, might improve this arrangement, if proper length is used.

17 Another disposition of nozzles is shown in Fig. 15. This may be called an arrangement in pairs, or in parallel. The pairs of cylinders which have the longest overlapping of exhaust, because of their crank-throw relations, should be connected to the same exhaust nozzles. The common discharge from these paired nozzles

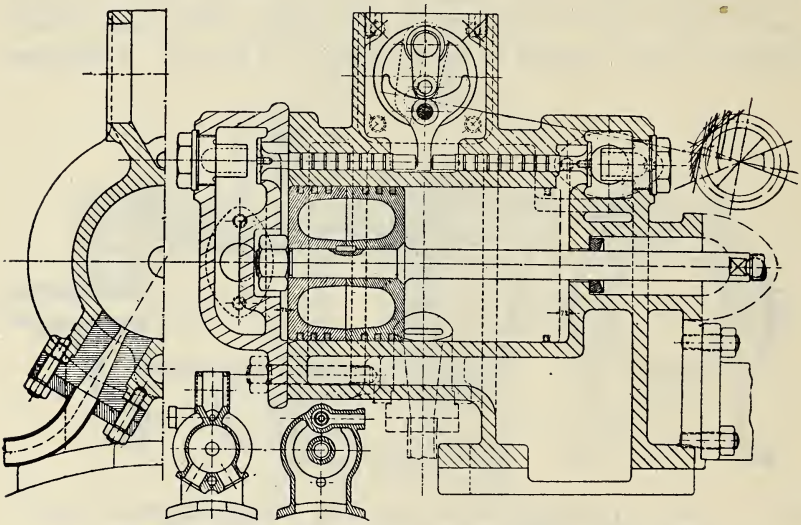


Fig. 12 New Design with Shorter Piston and Cylinder, Two Nozzles and No Compression Valves

should then be connected in other pairs, finally to discharge through a common exhaust pipe and diffuser.

18 If it is desired to take off exhaust steam at pressure above that of the atmosphere, the exhaust-pipe diffuser must be made part of a large receiver tank, from which a small pipe or pipes may lead steam to the heating system at nearly constant pressure. The volume of such a receiver must be large, or the changes of pressure in it will have too great a damping effect on the exhaust-pipe pulsations. It makes design by calculation still more difficult.

19 The practical application of long exhaust pipes, receivers, nozzles, etc., is by no means so difficult as their proper design. Experience with Humphrey pumping engines of all types and sizes has shown that the inertia effect of long pipes may be a decided

detriment to efficiency in some cases, but may be turned to decided advantage if properly utilized and controlled. One of the writers has seen the capacity of these internal-combustion engines im-

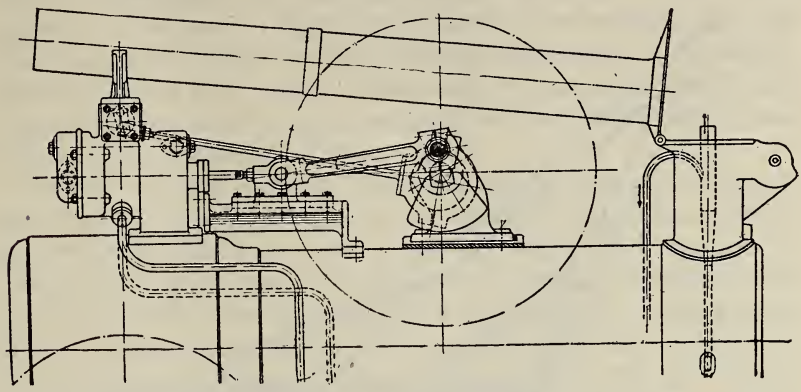


Fig. 13 Long Exhaust Pipes Leading to Smokepipe in New Design Cylinder

proved 50 per cent by simply changing the length of exhaust pipe. Fortunately the graphical methods of design developed for Humphrey pumping engines may be easily adapted to new uses. Ways must be found to take account of more variable factors (nine or ten

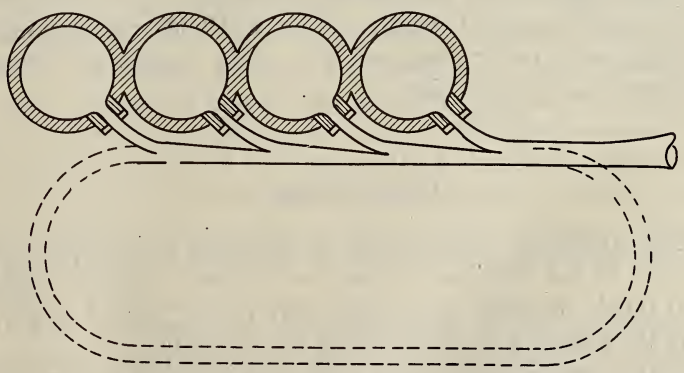


Fig. 14 Series Arrangement of Exhaust Pipes and Nozzles in Multi-Cylinder Engine

in this case); but that is easier by graphical than by any other method. It may, however, take some time and experimental work to determine the practical coefficients.

20 Long exhaust pipes and proper nozzles, however, do provide a means of controlling compression, which in a uniflow engine serves to improve thermal conditions, as well as mechanical cushioning, for which it is mostly used in ordinary steam engines. They

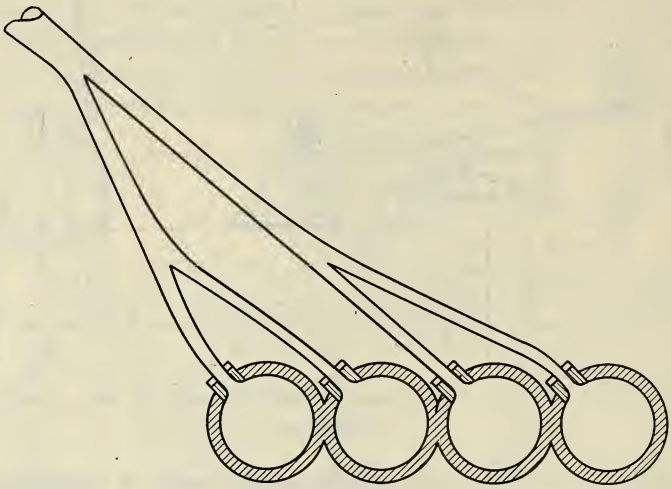


Fig. 15 Pair or Parallel Arrangement of Nozzles in Multi-Cylinder Engine

avoid large clearances in non-condensing engines, or the alternative of mechanically-operated exhaust valves, which the condensing uniflow engine has already dispensed with. The writers will welcome broad discussion of the subject.

DISCUSSION

D. B. Prentice. The Society is fortunate in hearing another paper, one of the authors of which is Professor Stumpf, who has done so much through the development of the uniflow engine to improve the economy of reciprocating steam engines.

We are so accustomed to the usual warnings that a reduction of back pressure in the cylinder of a reciprocating engine will, through the lower temperature, cause excessive cylinder condensation after the next admission, that we are apt to forget that with certain valve arrangements this does not follow; and the present paper clearly indicates a way by which a larger part of the available energy of

the steam may be converted into useful work without any apparent corresponding increase in the losses. The writer is sorry that the authors are not ready to present data from actual tests showing in percentages the increased power which can actually be secured. The paper suggests a fertile field for research and one which can be well cultivated in our college laboratories. The experiments would be quite within the capacity of graduate students and undoubtedly superior seniors could derive worthwhile results with comparatively simple apparatus.

If the per cent recovery approximates the values indicated in Fig. 2, a few simple calculations on adiabatic expansion indicate that the conversion from heat energy into useful work may be increased from 6½ to 8 per cent, the greater increase resulting if the steam is initially saturated and the lower increase resulting from the use of superheated steam. These calculations are based on ratios of expansion of approximately 4.8 to 1 which will give release pressures of 5 pounds above atmosphere for moderate steam pressures. Of course, these increases in conversion represent only very small improvement in thermal efficiency. Nevertheless, we are accustomed to consider that it is almost impossible to get any improvement in the performance of a well-designed uniflow engine.

Whether or not objectionable noises would be produced under the varying cut-offs required in an engine running under changing load conditions can be determined only by actual experiment, but it is probable that they could be eliminated under all conditions by changes in the shape, number, and location of the exhaust ports. The writer is doubtful whether the principles suggested in this paper will go far toward solving the well-known difficulties with the two-cycle explosive engine, as the problem there is probably more closely connected with the pressure and velocity conditions of the incoming charge than with the scavenging effect of the exhaust. However, experimentation along this line for this type of engine is undoubtedly desirable.

It is to be hoped that the presentation of this paper will provoke research along the lines suggested and that another year may produce exact experimental results which may well form the material for a valuable contribution to the papers of the Society.

J. D. Riggs. In Par. 3 we read: "It is probable that the studied application of suitable exhaust pipes with properly designed nozzles for gas and steam engines, especially those with piston-

controlled exhaust ports, is new, for a United States patent has just been allowed on it."

If there exists any one machine which has been studied from end to end, by all manner of men, that machine would seem to be the reciprocating steam engine and it is difficult to accept the above statement in its broadest sense. However, when restricted to the application of a principle, as all patents are, the writer is not disposed to question the statement.

While the writer was an engineering student some of us brought up the question of how much energy might be recovered from the exhaust pipe of the laboratory engine by putting on a diverging nozzle, and after a brief discussion we agreed that the quantity was so small as to be negligible. But if the exhaust had been turned into the heating system then something worth while might be recovered. The initial pressure was about 70 lb., expanding to about 7 at release in a small compound engine making 350 r.p.m. and developing 10 to 12 horsepower.

J. Stumpf. May I add some remarks about the uniflow locomotive mentioned by C. C. Trump, which was tested on several trial trips.

The ribs in the exhaust nozzle near the top shown in Fig. 10 of our paper were found to be superfluous.

The production of steam was ample, the fire, even when running under heavy load, could be thin. With medium load we found 12 per cent CO_2, no CO, and 50 per cent surplus air. With heavy load we found 13 per cent CO_2, some traces of CO, and 48 per cent surplus air. In other locomotives a thick fire must be maintained with a production of CO_2 in the lower part of the coal, and a reduction to CO in the upper part. This causes final combustion of CO into CO_2 in the firebox and in the tubes, with corresponding high temperatures in the smokebox.

The uniflow locomotive does away with the reduction and second combustion, gives complete effect of radiant heat in the firebox, and low temperature around the coils of the superheater. Consequently there is a good thermal efficiency but insufficient superheat.

Summary:

1 The grate area was too large
2 The boiler was too large
3 The superheater was too small.

The superheater was, therefore, enlarged by lengthening the coils toward the firebox.

The draft in the smokebox was $1\frac{1}{2}$ to 2 in. water column, in comparison with about 3 in. in ordinary locomotives.

The exhaust nozzle in the cylinder wall transfers the pressure energy of the toe of the diagram, with fairly good efficiency, into speed energy, which is used in the ejector nozzles, partly for drawing residual steam out of the other cylinder. High-speed steam always meets low-speed steam, thus producing mixture of steam with some equalization of momentum.

The diffusor will reduce the violence of the puffs, diminish the speed and equalize the flow of steam. This again requires an ample spreading of the steam with ample dimensions of the smokestack, but with a minimum exhaust loss. Ample and uniform withdrawal of the combustion gases, light draft in the smokebox and good combustion are secured. The light draft is also a consequence of the small depth of fire, and good combustion requires a relatively small volume of air.

The only difficulty with this locomotive reported by the railway authorities in Germany was reduced traction due to the variation of torque, as a result of high compression. A small piston exhaust valve is therefore being added to secure a more even pull at slow speeds and a better use of the adhesive weight.

With a two-cylinder locomotive there still remains uneven torque. This can be still further improved by replacing two cylinders, with cranks at 90 deg., by 3 cylinders with cranks at 120 deg. If these cylinders have equal displacement compared with the two cylinders the pull will be increased by about 15 per cent.

In my opinion this 15 per cent is a strong argument in favor of three-cylinder locomotives. In addition there is the gain in overlapping exhausts, as mentioned in the paper. With the two-cylinder locomotives on quite heavy loads the indicator cards showed a vacuum of nearly 15 in. in the exhaust pipe between the cylinder and the junction, without exhaust valves. Over 20 in. vacuum may be expected with exhaust valves on two cylinders and with or without them on three-cylinder locomotives. With proper area of exhaust nozzles a good part of this vacuum will penetrate into the cylinders. With the first locomotive on its first trials calculation of the compression indicated a vacuum of more than 4 in. of mercury.

Comparing this with a back pressure of 7 in. of mercury above atmosphere in an ordinary locomotive the net gain represents more than 11 in. of mercury.

The addition of piston exhaust valves not only improves the starting torque, but also makes it possible to reduce the clearance space in the cylinders from 12 to 7 per cent. The resulting gain in efficiency more than offsets the slight loss which may result from passing a small quantity of exhaust steam over hot clearance surfaces. The net result of these improvements is a marked reduction in fuel consumption.

The Russian Government has taken a great interest in this new uniflow locomotive which has now been in operation about one year, with great success. Special interest has been aroused by the new exhaust device which is against all tradition in locomotive practice. The Russians have ordered two of these locomotives with suction exhaust and piston valves combining inlet and additional exhaust. They are being built in Sweden and are to be used for ordinary freight service.

It is a matter of course that the new suction draft can be used to great advantage to internal combustion engines, in which the energy contained in the gases at release is usually larger than in steam engines. Consequently the mechanical vacuum, the load of mixture and the output must be much larger.

At present I am designing a six-cylinder motor with the new suction exhaust for the Benz Automobile Co. at Mannheim.

No. 1843

SOME NOTES ON RAILWAY REFRIGERATOR CARS

By W. H. Winterrowd, Montreal, Quebec, Canada.
Member of the Society

An efficient and economical railway refrigerator car is one which provides adequate air circulation, adequate protection to the lading, adequate quantity and degree of refrigeration, quick initial cooling, uniform temperature, dry air, space to permit proper methods of loading, and good car construction to minimize maintenance and increase time in service.

The literature of this subject being to some extent scattered, this paper is an attempt to include under one heading some of the most interesting and important facts regarding principles and methods involved in railway-refrigerator-car operation, as well as to present some information regarding the types of cars and methods of construction used by various railways and private-car owners.

The data were collected directly from the railways and private-car owners in an endeavor to sense the general trend of design and construction, and as there are signs on the horizon which indicate a renewed activity in the construction of railway equipment, this information is presented in the hope that it will serve to re-emphasize the great importance to the railways of efficient and well-maintained cars of the refrigerator type.

THE importance of efficient and well-maintained railway refrigerator cars to the people and railways of North America cannot be over-emphasized. The *Railway Equipment Register* for March, 1922, shows that the railways and private-car owners own approximately 153,000 cars of this type; figures from the same source indicate that this is about 6 per cent of all the cars owned, and approximately 13 per cent of all the box- or house-type cars listed.

2 Refrigerator cars are important in the life of the people because they are used principally to carry the bulk of the nation's perishable foodstuffs. They are important to the railways be-

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Presented at a Joint Meeting of the Metropolitan Section and the Railroad Division of the A. S. M. E., and the A. S. R. E., in New York, May 16, 1922.

cause they are factors in the production of revenue. These statements may appear trite, but their importance is such that the writer feels a re-emphasis is necessary and justifiable in any mention or discussion of this type of railway equipment.

3 The subject of railway refrigerator cars has been, and is still one of growing importance, a growth that has been contemporaneous and proportional to the development of the United States and Canada and their communities, and to the resultant demand and necessity of transporting increasing quantities of perishable commodities greater and greater distances. When hauls were short and the variety of perishable commodities few, the problem of transporting and protecting the commodities from heat or cold was comparatively simple. Increasing distances and a greater variety of perishables not only made necessary greater numbers of, and more efficient, cars, but involved the establishment of railway-divisional and terminal facilities, upon which the successful operation of this type of equipment is contingent.

LITERATURE ON THIS SUBJECT IS SCATTERED

4 A very interesting and valuable description of the development of the railway refrigerator car and its resulting economic importance is contained in two papers entitled The Refrigerator Car — Retrospective and Prospective, presented by E. F. McPike and J. H. Bracken, before the American Society of Refrigerating Engineers in 1913. Comparatively speaking, literature upon the subject is not vast, although exceedingly valuable. Until several years ago the available information consisted chiefly of descriptive articles in the technical press and papers presented before technical and engineering societies, including the International Congress of Refrigeration. In later years the subject became one of investigation by the U. S. Bureau of Agriculture, as well as to some extent by the Bureau of Agriculture of the Canadian Government. Bulletins issued by these departments and papers presented by Dr. M. E. Pennington, then Chief of the Research Laboratory, Bureau of Chemistry, U. S. Department of Agriculture, and the previously mentioned papers of Messrs. Bracken and McPike, are among the most important and valuable contributions to the subject.

5 Finding the sources of information to some extent scattered, the writer has attempted to include under one heading some of the most interesting and important facts regarding principles

and methods involved, as well as to present some information regarding the type of cars and methods of construction used by various railways and private-car owners.

PRESENT DATA GATHERED TO SENSE TREND IN ART

6 The data in this paper were collected directly from the railways and private-car owners in an endeavor to sense the general trend of design and construction, and as there are signs on the horizon which indicate a renewed activity in the construction of railway equipment, this information is presented in the hope that it will serve to re-emphasize the great importance to the railways of efficient and well-maintained cars of the refrigerator type.

7 The information which follows deals entirely with cars in which refrigeration is obtained by means of ice and salt. Many interesting efforts have been made to evolve a mechanical means of refrigeration, but the problem has been difficult of satisfactory and economical solution.

8 Traffic conditions have been and remain such that it is desirable to treat the railway refrigerator car as an individual unit. Interchange of equipment, the occasional necessity of setting out a car for repairs, the holding of cars for re-consignment, and individual movements involving scattered loading points and widely separated destinations have made the individual unit most economical.

9 Further reference to the subject is unnecessary because a discussion of the experiments, and the practicability as well as the possibilities of mechanical refrigeration applied to railway cars will be presented before this meeting by another author.

REFRIGERATION OF COMMODITIES IN TRANSIT INVOLVES MANY FACTORS

10 The important factors involved in the efficient refrigeration of commodities in transit, or in protecting them from cold, have been adequately described in the papers already cited, but it may be as well to re-emphasize them very briefly as a preface to the information which is to follow.

11 The prevailing method of obtaining refrigeration is by means of naturally-circulated air, cooled by contact with ice, or ice and salt, placed in suitable receptacles called bunkers located

at each end of an insulated car. Some modifications of this system will be touched upon very briefly later.

12 Circulation is assisted and made most efficient by means of insulated partitions, called bulkheads, placed in front of the containers and so constructed that the relatively warm air must pass over the top of them to reach the ice, or ice containers. The air becoming chilled, and therefore heavier, sinks toward the floor and reaches the body of the car by passing through a space beneath the bulkheads.

13 These insulated partitions also assist in protecting the lading nearest the ice containers. Without bulkheads, and when salt is used with the ice to hasten and increase refrigeration, that part

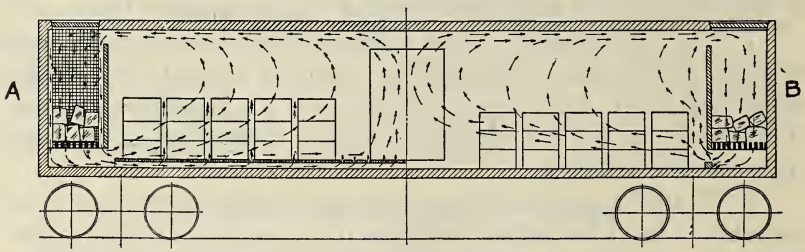

Fig. 1 Diagram of Air Circulation

of the lading nearest the ice frequently freezes, an undesirable and disastrous occurrence with some commodities. At the same time, that portion of the load near the center and top of the car may remain at too high a temperature, an equally undesirable condition.

14 As a further aid to circulation, particularly in cars where the lading is piled or stacked, a slatted wooden structure known as a floor rack is of very great value. These racks, on the top of which the lading may be placed, consist of longitudinal runners 3 or 4 in. high, with cross slats fastened to the top of them. They are hinged to the side walls of the car and are divided in the middle so that they can be turned up to make the floor accessible. These racks permit the cold air which flows beneath the bulkhead to circulate freely toward the center of the car and up through the lading.

ILLUSTRATIONS OF EFFECT OF DESIGN

15 The " A " end of the car, Fig. 1, shows the relative arrangement of bunkers, bulkheads, and floor racks, and the resultant trend of air circulation.

16 It is highly desirable that there be no obstruction to the

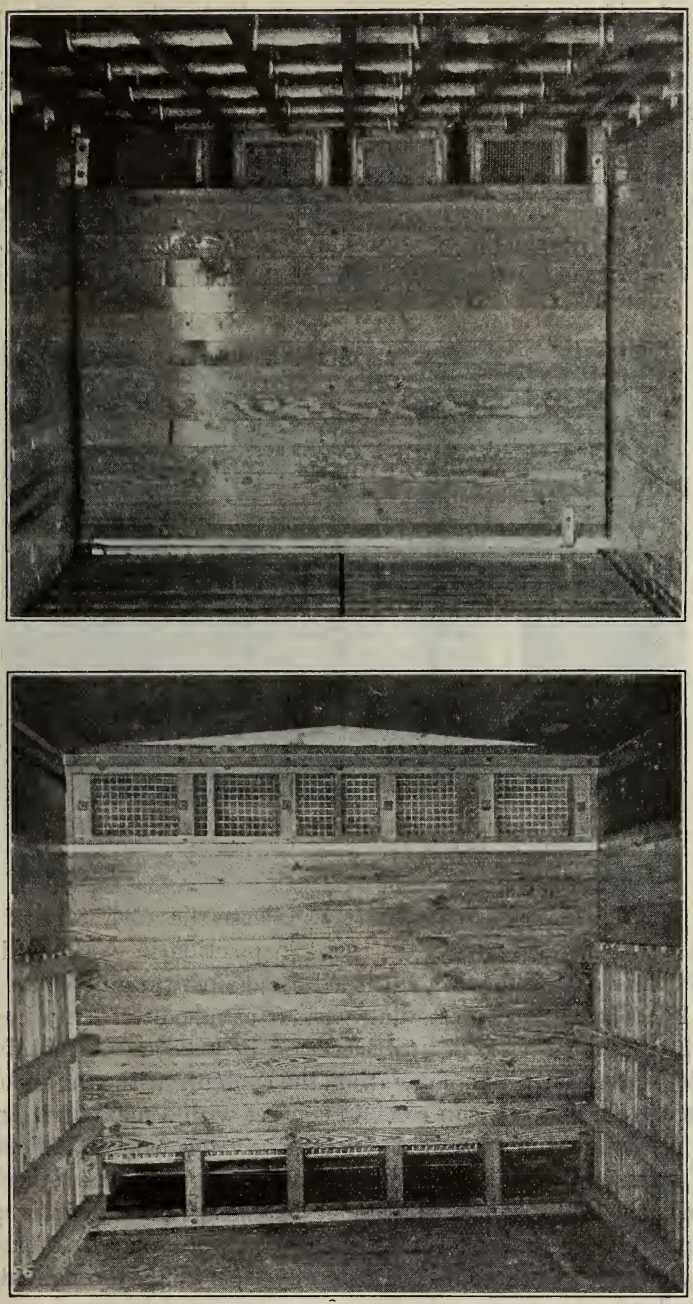

Fig. 2 Floor Racks in Refrigerator Car

flow of cold air where it passes beneath the bulkheads. The "B" end of the car, Fig. 1, shows how an obstruction at that point can act as a dam or deflector of the air currents, and partially or entirely defeat the object of the floor racks, and even cause frosting of the lading against the insulated bulkhead. If floor, bunker, or splash-board construction necessitates a ledge beneath the bulkhead, it should be kept as low as possible and the floor rack and bulkhead so designed to provide sufficient area for a free flow of air.

17 At the "B" end the lading is shown piled directly on the

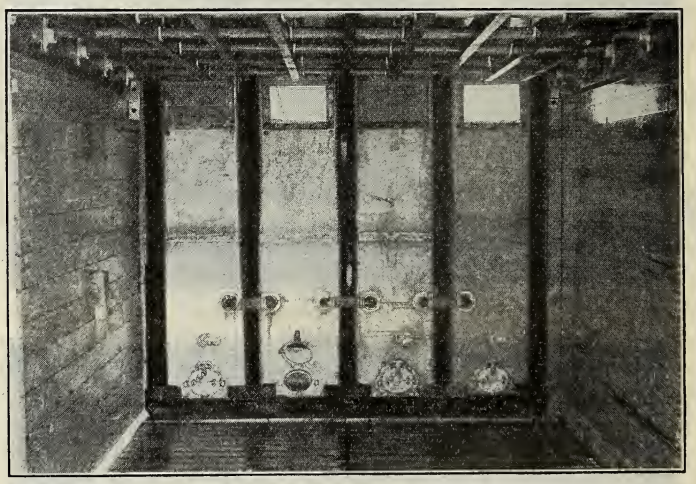

Fig. 3 Brine Tanks in Refrigerator Car

floor and the air currents indirectly indicate the advantage of floor racks. Fig. 2 illustrates these racks in position in a car, and also shows their appearance when turned up against the side walls.

18 The method of loading various commodities is an important factor in their proper refrigeration in transit. To obtain the greatest advantage from circulation, the contents of the car should be loaded so that the air can come in contact with a maximum surface with a minimum of restricted circulation. It is easy to understand that boxes or containers, placed closely against each other and against the walls of the car cannot be cooled quickly or properly preserved at as uniform a temperature as when placed so that air can flow between them, or generally speaking, throughout the entire load.

19 The temperature of the circulating air is affected by the type and size of ice containers or bunkers. The chief considerations in the construction and capacity of the bunkers are the refrigeration required to replace the loss due to transmission and the refrigeration required to cool the car quickly and maintain its contents at the required temperature. The loss in transmission in turn depends upon insulation, car construction and maintenance; factors which will be more fully discussed later.

20 A basket bunker is shown in the "A" end of the car in Fig. 1, and a box bunker in the "B" end. Brine tanks are shown in Fig. 3.

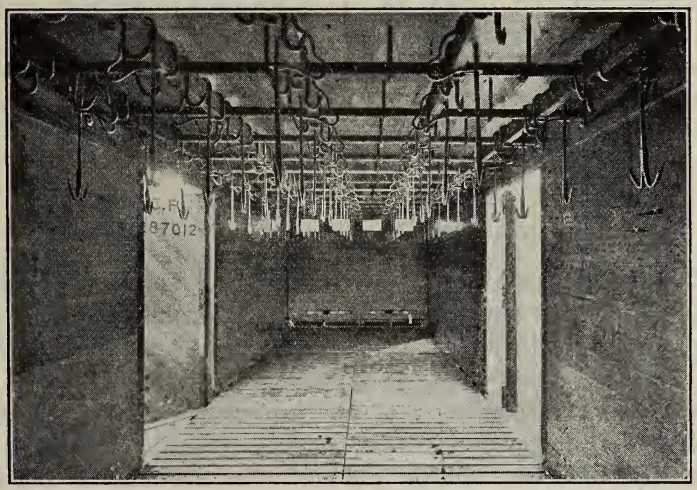

Fig. 4 Meat Rack and Hook Installation

21 The sides of the basket bunker are constructed of wire mesh. The bottom consists of a slatted wooden structure. The bunker is placed in position so that air space exists around the outer surfaces, this construction permitting the air behind the bulkhead to come in contact with a maximum ice area, while the open spaces around the bunkers facilitate circulation.

22 The bottom of the box bunker shown at the "B" end of the car in Fig. 1 is also slatted. The walls are formed by the bulkhead and by the sides and end of the car. In an endeavor to make the refrigerator car more productive and useful for general purposes, some builders have applied collapsible box bunkers. In

this design, the bulkhead is swung up, or to one side, and fastened. The slatted rack is swung or folded back against the end of the car. In this way the space occupied by the bunker is made available for general lading.

23 Another very important factor is the size of the car and its proportions, particularly with reference to the distance between the bulkheads. The cold air passing out beneath these insulated partitions flows toward the center of the car. In doing its work it gathers heat and grows warmer. If the distance between bulkheads is too great, the air may not be at a low enough temperature when it reaches the center of the car to properly refrigerate the load at that point, particularly near the top of the car. Under such conditions a portion of the load near the bulkheads may be sufficiently chilled while the upper and center part of the load may be too warm. The chief factors generally considered in this connection are type and capacity of car, type of bunkers, amount of commodities and size of containers, and temperature required for efficient refrigeration.

IMPORTANCE OF TEMPERATURE OF ENTERING LOAD

24 The temperature of the entering load is a very important factor. If heat is not removed from certain commodities prior to loading it is highly desirable to remove it as quickly as possible after loading. Tests and general experience show that if the heat is not promptly reduced, the commodities either spoil en route or reach their destination in such condition that their market value is greatly reduced. Quick cooling after loading is generally attempted by pre-cooling the car, by the use of cars in which maximum and most efficient air circulation can be obtained and by mixing a proper amount of salt with the ice or by placing coarse rock salt on top of the ice.

25 When salt is mixed with the ice, the greatest amount is used at the first icing, the time when the greatest quantity of heat is in the lading. At the second icing the amount is generally reduced, while at the third and successive icings a stipulated minimum amount is used.

26 Pre-cooling the car may be accomplished by the use of ice and salt, but at many points where a large tonnage of fruit or meat originate, the cars as well as the lading are often pre-cooled by mechanical means.

27 Pre-cooling means less ice in transit, a matter of economy to shippers and railways alike. Some commodities can be frozen hard and therefore require little or no icing en route, the lading itself supplying the necessary refrigeration. This not only insures better condition in transit but is an added economy.

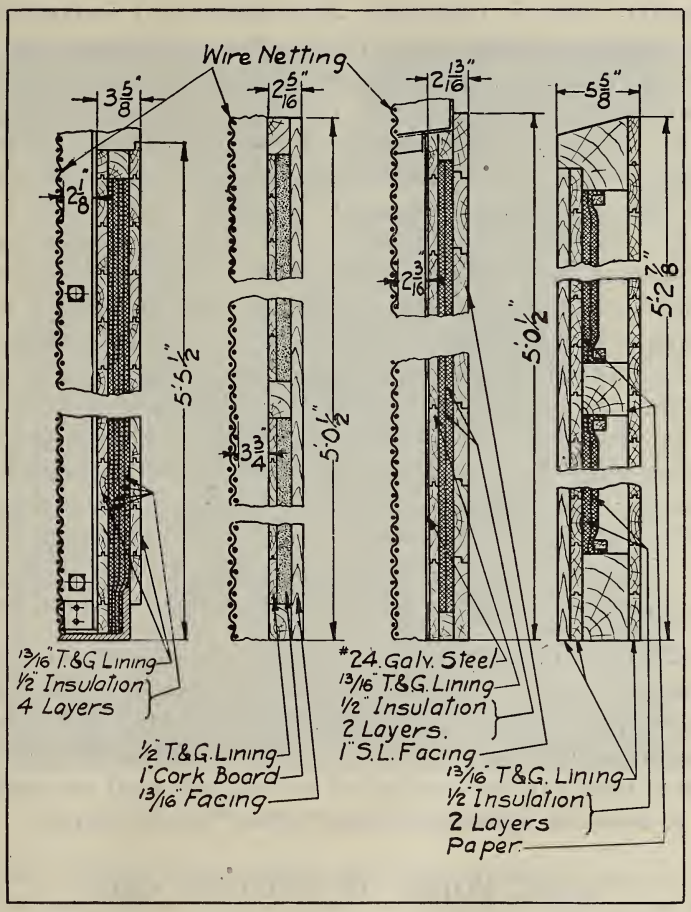

Fig. 5 Cross Sections of Bulkheads

28 Humidity or moisture content of the air in the car is almost as important as temperature. Generally speaking, if refrigeration is effective and a high initial rate of cooling obtained, the air is kept sufficiently dry due to condensation taking place on the surface of the ice, or ice containers. In this way the moisture given off

by some classes of lading is also deposited. Excess of humidity, if not fatal, is highly injurious to many commodities.

INSULATION IS THE MOST IMPORTANT CONSIDERATION

29 Finally we come to the matter of insulation, the most important factor in connection with efficient and economical refrigeration in a railway car. The function of the insulation is to afford protection to the lading by minimizing heat transmission through the walls, roof, and floor of the car. To do its work properly it must be by nature a poor conductor of heat. Other desirable qualifications are reasonable cost, strength, adaptability, durability, light weight, and imperviousness to moisture.

30 This subject might well be divided into two parts, insulating materials and insulation, because the general subject involves methods of car construction and maintenance in addition to the consideration of materials and principles of heat transmission.

31 The whole subject is capable of analysis but that the principles involved are not broadly known is indicated by the fact that in very recent years cars have been built without proper or sufficient insulation. These cars which do not properly protect their lading are huge consumers of ice and expensive to operate and require a great deal of maintenance to keep them in service.

SUMMATION OF REQUIREMENTS FOR EFFICIENT REFRIGERATION

32 It is evident that an efficient and economical railway refrigerator car is one which provides adequate air circulation, adequate protection to the lading, adequate quantity and degree of refrigeration, quick initial cooling, uniform temperatures, dry air, space to permit proper methods of loading, and good car construction to minimize maintenance and increase time in service.

SOME NOTES ON EXISTING CARS

33 In an endeavor to sense the trend of refrigerator-car design, proportions, and construction, the writer addressed an inquiry to a number of railways and private-car owners. The replies, while not complete, were very generous. A comparison of the most interesting returns is shown in Table 1, and makes a very interesting study, although in any consideration of this table the fact must not be overlooked that possibly some of the railroads or owners, if building equipment today, might modify their designs.

Every road or owner represented owns at least one thousand cars. As

far as possible the cars shown were chosen from quantities built in comparatively recent years. Many of the old timers, really not refrigerator cars at all, were omitted. Even so, some of the cars built in recent years provoke question.

34 Another point is that during the past three years car building or re-building has been at a minimum. Even so, it is of great interest to note

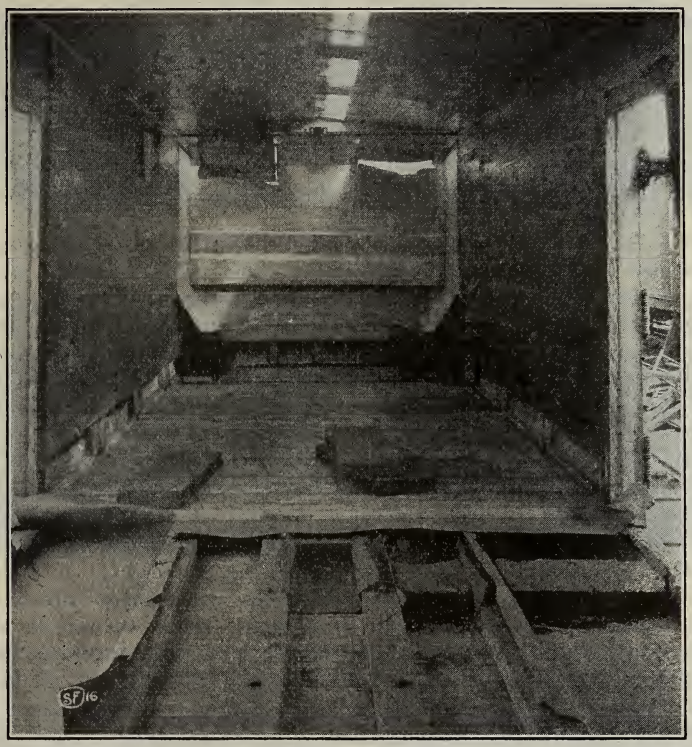

Fig. 6 Manner of Applying Insulation

that many of the cars built within this period, or being designed or constructed today, embody in great measure those principles which make for an efficient and economical unit.

TYPES OF CARS AND ICE CONTAINERS

35 Generally speaking the cars can be divided into two types: one, equipped with brine tanks and generally used for carrying meats; the other, equipped with bunkers, and used principally for carrying commodities such as eggs, butter, vegetables and fruit.

36 In connection with this distinction, based on ice containers, it is interesting to note that Dr. Pennington has stated that a car of the basket-bunker type, such as the U. S. Railway Administration Standard, will carry meat hung from rails quite as successfully as a car built especially for meat. The statement is also made that there is not visible in practical results the advantages supposed to accrue from the retention of the brine, provided coarse rock salt is placed on top of the ice in the bunker and so forced to bore its way through the whole mass before finding an exit.

37 But there is a very important problem in this connection that must not be overlooked if salt is to be used with ice in a basket bunker, and that is the method of disposing of the brine.

38 The subject of brine drip is one that has received a great deal of attention by railway engineers. It is common knowledge that if brine falls on journal boxes, side frames, arch bars or other truck parts, as well as upon rails, tie plates, bridge members, etc., the resulting damage is great and a factor involving heavy maintenance cost.

39 The subject is so important that the American Railway Association interchange rules specify that after July 1, 1922, no car carrying products which require for their refrigeration the use of ice and salt and which are equipped with brine tanks, will be accepted in interchange unless provided with a suitable device for retaining the brine between the icing stations.

40 If salt is to be used with ice in basket bunkers, a practical and economical arrangement is necessary to retain the brine so that it can be disposed of between icing stations.

41 The data submitted do not show any car of the basket bunker type equipped with overhead meat racks. They show that cars built for carrying meats and products requiring a low temperature are equipped with brine tanks.

42 Twenty-seven railroads and owners are represented in Table 1. Out of this number the principal cars of sixteen are equipped with bunkers, and the remainder are equipped with brine tanks. Out of the sixteen, eleven or practically 69 per cent are of the basket type; the remaining five, or approximately 31 per cent, are of the box type. The majority of the cars recently built, or now under construction are equipped with the basket type of bunker. The demand for refrigeration and the special-service car, as well as greater efficiency of the permanent basket type, appear to be decreasing the demand for the collapsible bunker.

43 Another distinction that prevails and will prevail as long as cars are built for some particular service is the difference in construction due to the commodity to be transported.

44 In meat cars the lading is hung on hooks suspended from a meat rack placed just below the ceiling. This rack generally consists of stringers and cross bars supported by the roof and walls of the car. In a car of this type it is necessary to make the framing heavier so that in addition to its other functions, it will adequately support the weight of the lading. A meat rack and hook installation is illustrated in Fig. 4. Very often additional lading is stowed or placed on the floor racks beneath the suspended load, in order to obtain the full carrying capacity of the car.

45 *Bulkheads.*—The majority of the cars tabulated are equipped with solid bulkheads. These are either built into place or are hinged from the walls or ceiling so that they can be swung open.

46 A few cars are equipped with the syphon system, in which the

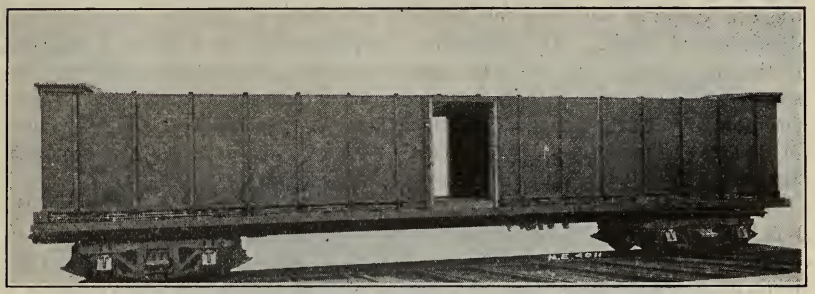

Fig. 7 Manner of Applying Insulation

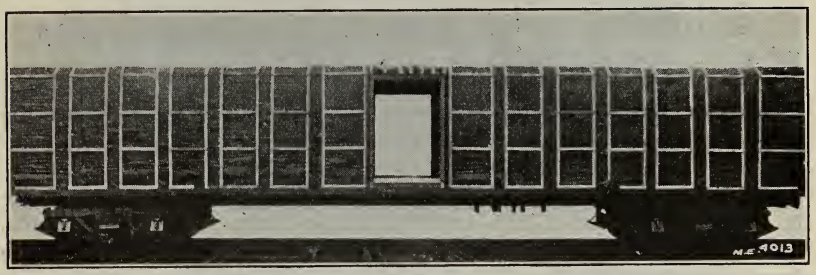

Fig. 8 Manner of Applying Insulation

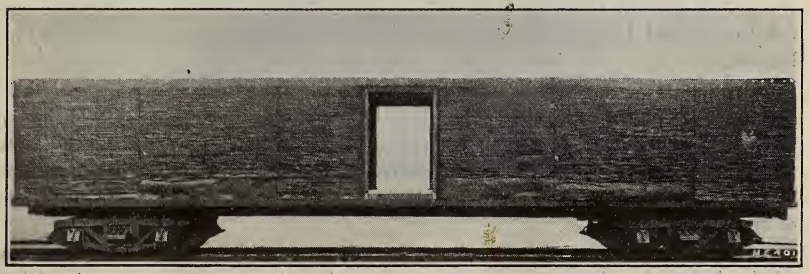

Fig. 9 Manner of Applying Insulation

bulkhead consists of a framework holding a series of galvanized iron louvres supposed to direct the air back and down into the bunkers. The theory is that air entering the bunker over the top of the bulkhead becomes chilled, and in its downward motion creates a suction or siphoning effect which draws air from the body of the car into the bunker through the openings

in the bulkhead. Although this system is on some cars of fairly recent origin it is significant that many railroads or owners who used it on their older cars have abandoned it in favor of the solid bulkhead.

47 The prevailing trend of construction indicates a recognition of the value of solid and insulated bulkheads. The general trend seems to be to use two layers of ½-in. hair felt between two walls of 13/16-in. matched- and dressed-wood lining. An interesting exception, and on a quite recent car, is the use of one layer of 1-in. cork insulation. Some bulkheads are constructed of two walls with a few layers of waterproof paper between them. Occasionally an air space is contained between the walls. In one instance, in addition to a dead-air space, two layers of ½-in. hair felt are provided. Some cross sections of bulkheads are shown in Fig. 5.

48 *Space Below Bulkheads.*—The space between the bottom of the bulkhead and the car floor varies considerably, ranging from 7 in. to 2 ft. 7 in. On the car with the 7-in. space the bulkhead is brought right down to the level of the floor rack. On the bottom of this bulkhead a canvas strip is fastened to prevent cold air passing out above the racks. The cars with the very large openings at both bottom and top of bulkheads are generally used for meat shipments.

49 The majority of the cars have a bottom opening of from 9 to 15¾ in. The average is about 12 in.

50 As the floor racks on these cars average about 4½ to 5 in. in height it can be seen that the cold air has access to the body of the car above the rack as well as through the space beneath it.

51 The writer has endeavored to ascertain if there is any relation between the size of the openings above and below the bulkhead and the velocity of the air in circulation, but inquiry has not produced anything definite. There is some unanimity of opinion, however, in favor of the design in which there is an opening of from 2 to 7 in. above the floor racks.

52 If there are any data on this subject they should be of considerable interest, but if no definite information exists the matter appears to be one worthy of careful investigation. It is evident that any improvement in circulation would help to bring about more uniform temperatures, an obvious benefit.

53 In the matter of efficient refrigeration the distance between bulkheads is an important one. The tabulation shows that this varies between 28 ft. 8 in. and 38 ft. 10 in. The general trend is between 32 and 34 ft. On the latest cars the spacing is approximately 33 ft., or slightly greater. The size of the standard egg crate has been a large factor in the establishment of the exact dimension.

54 Difficulty in obtaining proper temperatures at the center and top of the lading has been responsible for the thought that longer cars and less deep loading would bring better results. Longer cars have been demanded also as the result of a desire to increase their capacity.

55 The principle has been emphasized that heat transmission varies directly as the number of square feet of surface enclosing the car space. A study of some of the long cars indicates that this principle has not been followed closely in determining the kind and amount of insulation.

56 *Floor Racks.*—Space between the top of the floor rack and the floor of the car averages between 4½ and 5 in. The majority of the modern cars are equipped with these racks, but an examination of the tabulation would indicate that their importance is not fully recognized. This fact is borne out by an examination of hundreds of refrigerator cars at a fruit- and produce-distributing station. Many of the cars with long slats or runners

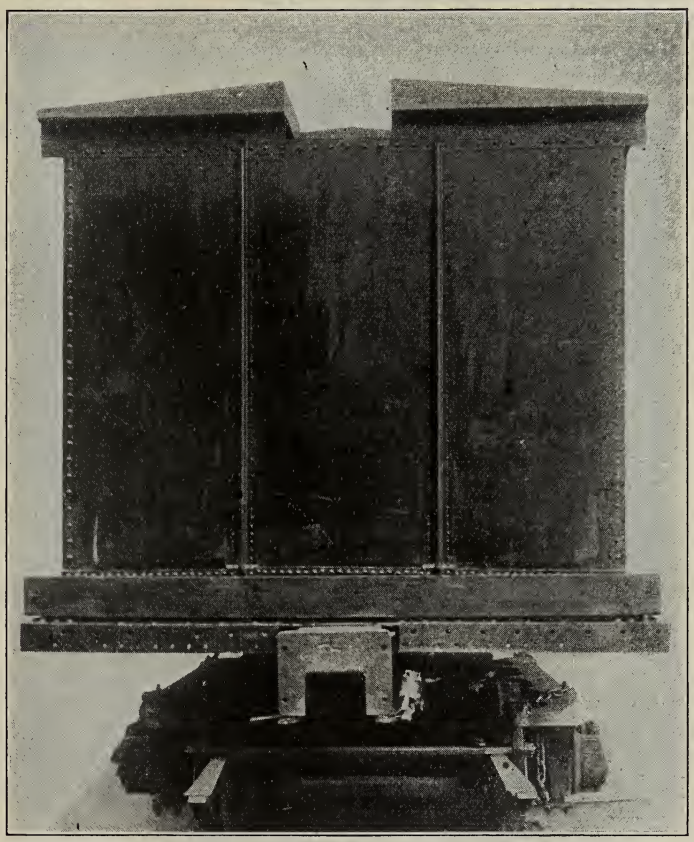

Fig. 10 Manner of Applying Insulation

fastened to the floor are of such construction and equipped with such types of bulkheads that floor racks could be applied easily and cheapily.

CAR CONSTRUCTION AND MAINTENANCE

57 An impression seems to prevail that the life of a railway refrigerator car is about 6 to 8 years. In 1919 a committee of the Mechanical Section of the American Railway Association reported that the average life of railroad-owned wooden refrigerator cars, dismantled, was 17.1 years, and of

private-line wooden refrigerator cars, dismantled, 21.9 years, making the average life for all wooden refrigerator cars, dismantled, 19.4 years. It was also stated that the average life of railroad-owned wooden refrigerator cars was largely affected by two lines reporting the dismantling of a large number of cars of an average life of only 15 years; by excluding these two lots of cars, the average life for railroad-owned wooden refrigerator cars was 21.3 years, and for private-line-owned cars 21.9 years.

58 The life of refrigerator cars equipped with steel under-frames or steel framing and superstructure is a matter upon which there are little data, because such cars are comparatively modern. There seem to be no reasons, however, barring those of possible evolution, why such cars should not have a long life and require little for maintenance by reason of their better design and construction.

59 It is not difficult to appreciate the causes responsible for the high cost of maintenance of old wooden cars; the refrigerator type does not stand alone in this class. But in addition to more severe traffic conditions, this type of car has required attention on account of the difficulty in keeping moisture away from the insulation as well as from the wooden framing and flooring. If the insulation becomes broken, wet or sags so that air can circulate around it, the car rapidly loses its efficiency. Table 1, pp. 38–39, and the cross sections in Figs. 14 to 24 inclusive, give a general idea of some types of cars, and what has been done to improve design and construction. Figs. 23 and 24 represent cars of relatively low efficiency. Figs. 14 to 22 inclusive show more modern cars and indicate the more recent trend in the matter of improved insulation and general construction.

60 The cross sections really speak for themselves, but a brief discussion under the separate headings Floors, Walls and Roofs will be some interest.

Floors.—The chief problem in floor construction is to make the structure waterproof, as well as a good insulator. Moisture and water finding its way through the floor or along the floor boards into the walls of the car, have been responsible for much trouble and expense.

61 The insulating value of all materials that absorb moisture is greatly decreased when water is absorbed. In addition, water causes most of the insulating materials popular in refrigerator construction to become mushy and sag or drop out of place. It also causes wood floors, lining and framing to decay and weaken, thereby making it more difficult to keep the general structure tight.

62 Nearly all the modern or at least recently built floors employ a construction involving cork as an insulating material. With two exceptions, shown in Figs. 17 and 18, the cork is applied in one layer. To keep moisture away from the cork various waterproofing compounds or waterproof materials are used. Fig. 6 shows a photograph of the floor and manner of applying insulation cross-sectioned in Fig. 16.

63 The one exception to this general trend is shown in Fig. 22, where the insulation consists of four massed layers of $\frac{1}{2}$-in. hair felt. Moisture is kept away from the top of the insulation by means of two layers of floor boards between which is laid a layer of waterproofing compound. The surface of the top floor is covered with a layer of waterproofing compound into the surface of which sand has been rolled.

64 Figs 23 and 24 show a floor insulation with intervening dead-air spaces. In past years it was the opinion that this type of construction gave the highest insulation value in walls and roofs as well as in floors. More recent opinion differs because experience has shown that unless unusual methods of construction or maintenance are used, it is very difficult to keep the air spaces tight. To be insulators, they must be dead-air spaces; once circulation starts their efficiency is destroyed.

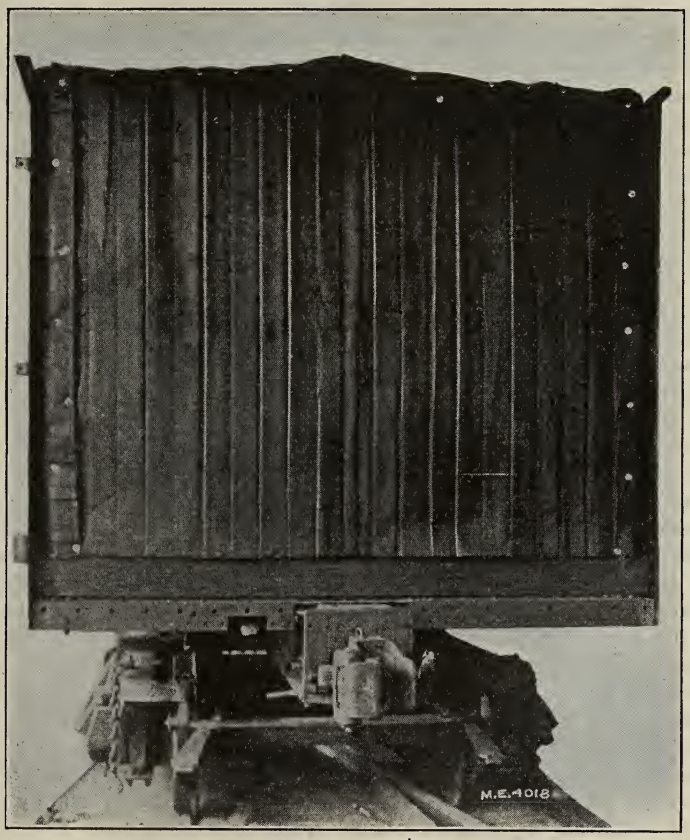

Fig. 11 Manner of Applying Insulation

65 It has been intimated that in some cases cork as a floor insulator has not been entirely satisfactory because in time it becomes brittle and crumbles. Specific information on this subject would be very valuable, as it would indicate whether the trouble was inherent or due to some particular method of construction.

66 *Walls.*—In connection with a waterproof structure, it is interesting to note the various methods employed at the junction of the floor and side

walls to keep water from getting past the lining and into the insulation. This point has been a source of great trouble. Some particularly interesting methods of construction at this point are shown in Figs. 16, 20, 21, and 22.

67 An exceedingly interesting example of waterproof construction is contained in some all-steel refrigerator cars designed by W. F. Keisel, Mem. Am. Soc. M. E., Mechanical Engineer of the Pennsylvania Railroad. The proportions of these cars were described in the *Railway Review* of February 3, 1917. An interesting feature in connection with these cars is the fact that the body of the car consists of an all-steel container placed within an outer container, the space between the walls being filled with insulation. The writer understands that at the floor, the sections of the inner container are welded together, thus making the floor practically one piece and water-tight and thereby affording maximum protection to the insulation. Figs. 7 to 12 inclusive illustrate the inner and outer container, and the methods of applying the insulation.

68 Inspection of the various cross sections indicates a general trend toward massing wall insulation and eliminating air spaces between the layers of insulation. As a rule the insulation is applied in a continuous strip from door post to door post. The advantage of applying insulation in this way lies in the fact that a continuous or unbroken surface presents no joints or openings through which air can pass or circulate. It has been the experience that where insulation is applied in sections, unusual construction is required to prevent eventual air circulation. Wall insulation is rarely less than 2 in. thick on the most recent cars. In some cases this insulation is applied in two massed layers. In one case the single layer is 2-in. thick. In the majority of cases four massed layers of ½-in. insulation are used.

69 The construction employed by the U. S. Railway Administration is indicated in Fig. 14 and shows the insulating material massed beneath the outside sheathing. Air space is provided between the inner lining and the blind lining. This construction was advocated as a method of preventing damage to the insulation should nails be driven through the inside lining.

70 A great many cars are insulated in this way, but there are some interesting exceptions, one of which is shown in Fig. 17. The advantage claimed for such construction is that if a car becomes cornered or damaged to such an extent that sheathing is cut or broken, the insulation stands a much better chance of remaining intact or becoming only slightly damaged, and the lading not subjected to risk caused by loss of cold air. It is also claimed that by the use of properly constructed wood forms or spacers, and by proper loading methods, no necessity should exist for driving nails through the inside lining. A great many railroads are conducting an educational campaign in this connection.

71 Two interesting wall structures are shown in Figs. 17 and 18. In these figures it will be seen that layers of cork board are used below the insulation, the hair felt starting at a point 13 in. above the level of the floor.

72 Another arrangement in which cork is used is shown in Fig. 21. In this cross section the two inner and massed layers of hair-felt insulations come right down to the floor level. A slab of cork board is placed between the hair felt and the inner lining. The hair felt is protected at the floor line

with canvas duck laid in white lead. The bottom of the insulation behind the sheathing is protected by waterproof compound poured into a channel or gutter provided for the purpose. The compound in this gutter joins the compound laid above the flooring cork thus forming a continuous waterproof section.

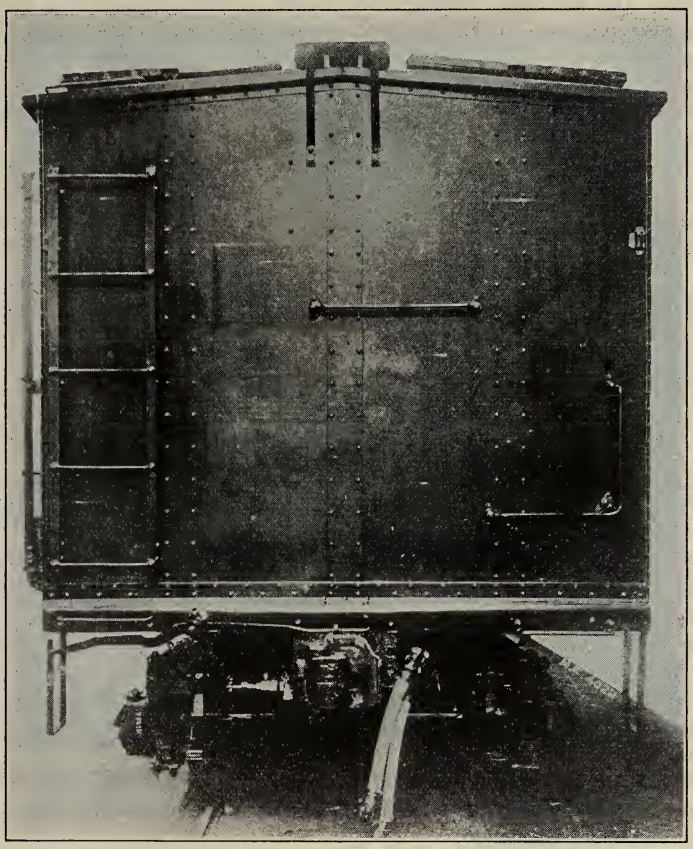

Fig. 12 Manner of Applying Insulation

73 *Roofs.*—The tendency is to apply massed insulation in the roofs. As a rule the most modern cars have 2 to 2½ in. of insulation applied in this way. The car shown in Fig. 17 has 3 in. of such insulation.

74 Some cars are equipped with a carefully designed double-board roof with waterproofing compound between the layers. There are many advocates of this type of roof, but it is interesting to note the number of outside-metal roofs that are applied to cars of this type. The advocates of the outside-metal roof claim a saving in weight and greater protection to

the sub-roofs and insulation from moisture, claiming that with proper insulation the metal roof has no effect on the interior temperature of the car.

MISCELLANEOUS

75 *Doors and Hatches.*—Doors and hatches are being made with more insulation and are being strongly and properly constructed so that they will fit the door openings tightly, and not permit any loss of refrigeration due to leakage. In this connection, any other openings into the car should be so constructed that they can be kept tightly in place and easily maintained. An efficient door-locking device is no small item in keeping doors tight, and thereby maintaining the efficiency of the car.

76 *Painting.*—Refrigerator cars should be kept well painted in order to preserve all exterior surfaces. This is in the interest of obtaining long life for the car. Metal parts should be given particular attention in this respect.

77 The writer believes that refrigerator cars should be painted with a light or non-heat absorbing color. Dark colors absorb heat. An inquiry addressed to the owners of white and yellow cars indicated that no specific data existed on the subject, but it was the general belief that the light colors were an advantage in this respect. Accurate information on this subject would be of great interest.

INSULATION

78 The previous references to insulation have been very brief, and have referred principally to its use in general car construction. The following paragraphs contain some notes regarding insulating materials.

79 It has been stated that the function of the insulation is to afford protection to the contents of the car by minimizing heat transmission through the walls, roof, and floor. A good insulator must not only be a poor conductor of heat, but must be a material having the qualifications of reasonable cost, adaptibility, durability, light weight, imperviousness to moisture, freedom from odors, and be proof against vermin. In any study of insulating materials for use in railway refrigerator cars, these factors must all be kept in mind.

80 There are some materials that have a low thermal conductivity, but their other qualifications make them unsuitable for practical use in the type of car under discussion.

81 A good insulation, but one which adds greatly to the weight of the car is undesirable. In the interest of economical transportation the car should be no heavier than is necessary to obtain the required efficiency and the required strength to insure continuity of service and low cost of maintenance.

82 A material with low thermal conductivity, but one which is difficult to apply economically, is also undesirable. On the other hand, there may be materials easy to apply but which will not stay in place or which will not retain their insulating value under service conditions; these are equally undesirable.

83 In addition, the material should be of a kind easily handled as well as easily applied. Materials difficult to handle and difficult to apply

add unnecessarily to the first cost of the car, as well as to possible future maintenance.

84 It seems to be generally conceded that the best insulating materials are those which contain a very great number of minute dead-air cells, or interstices containing dead air. If these air cells become filled with moisture, the thermal conductivity of the material is increased. This is one of the

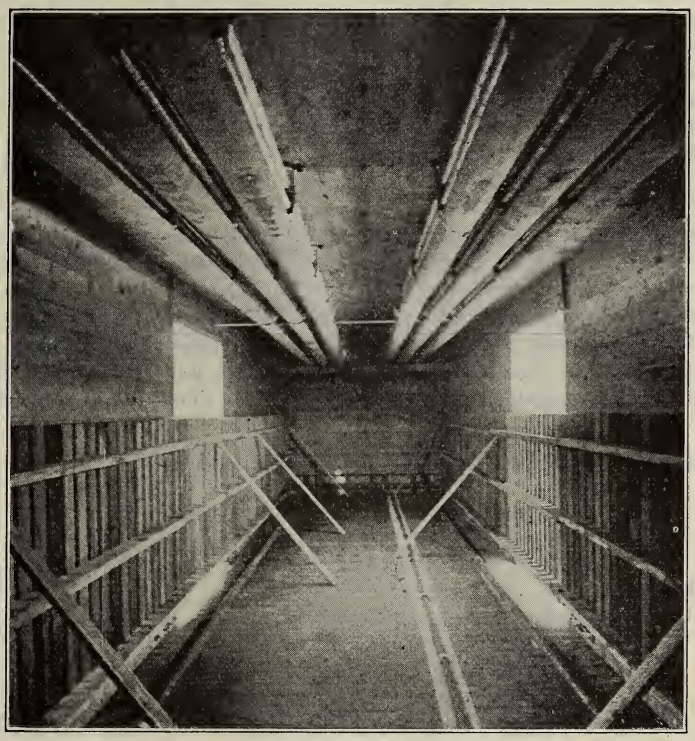

Fig. 13 Interior of a Car With Overhead Brine Tanks, Showing also Heater Pipes on Floor

reasons why it is so important to protect insulation from water, and why it is desirable to use a material highly resistant to moisture.

85 Some materials when subjected to moisture fall out of place or sag, and if a large air space, or pocket, is not formed, air circulation frequently results, the effect of which greatly decreases the efficiency of the car.

86 No argument is necessary to indicate the advantage and value of insulating material free from odors and vermin-proof.

87 *Thermal Conductivity.*—The question of thermal conductivity of various materials and compound structures is one that has been a matter of study and investigation for many years. The greatest part of the information available upon the subject is the result of tedious, exact and difficult experimentation.

88 The subject of heat transmission through the walls of a railway refrigerator car is one upon which there is some difference of opinion, this difference dealing largely with variables or factors which have not yet been reduced to absolute terms.

89 It is not within the scope of this paper to elaborate upon these differences or to enter into a discussion of all the factors involved. The most important of these deal with the effect of air velocity and moisture upon surface transmission, the effect of moisture upon conduction, the effect of radiation as a corrective factor, and the importance of knowing the condition of the general structure in order to determine the value of the so-called dead-air spaces. The bibliography at the end of the paper refers to literature in which these subjects are discussed in some detail.

90 In calculating heat transmission through a compound wall, it is essential to know the thermal conductivity of the various materials contained in the structure.

91 The most recent determinations of thermal value of various materials is shown in Table 2, p. 40, taken from the very interesting paper, The Thermal Conductivity of Heat Insulators, by M. S. Van Dusen, in the October, 1920, journal of the American Society of Heating and Ventilating Engineers.

92 The formula used for calculating heat transmission through a compound wall can be credited to Jean Claude Peclet, and is derived from his original formula contained in his Treatise on Heat (Traite de la Chaleur).

93 The formula for the total heat transmission is

$$H = \frac{1}{\frac{1}{K_1} + \frac{D}{C} + \frac{D_1}{C_1} + \frac{D_2}{C_2} + \text{etc.} + \frac{1}{K_2}} \quad \ldots\ldots\ldots \quad [1]$$

in which

H = Heat transmitted per sq. ft. per hr., b.t.u.
K_1 = Inner surface conduction
K_2 = Outer surface conduction
D, D_1, D_2, etc. = Thickness of each element in wall, in.
C, C_1, C_2, etc. = Thermal conductivity (per hour) of elements corresponding to thickness D, D_1, D_2, etc., per in. thickness per sq. ft. per deg. fahr.

94 If the difference between inside T_1 and outside T_2 temperatures is considered, Equation (1) should be multiplied by $(T_1 - T_2)$ and becomes

$$H = \frac{T_1 - T_2}{\frac{1}{K_1} + \frac{D}{C} + \frac{D_1}{C_1} + \frac{D_2}{C_2} + \text{etc.} + \frac{1}{K_2}}$$

95 The thermal conductivity of each material shown in Table 2 is stated in b.t.u. per day.

96 In the formula, thermal conductivity C, C_1, C_2, etc., is stated in b.t.u. per hour. To be used in the formula each thermal conductivity figure in the table must be divided by 24 to bring it to the required hourly basis. For example

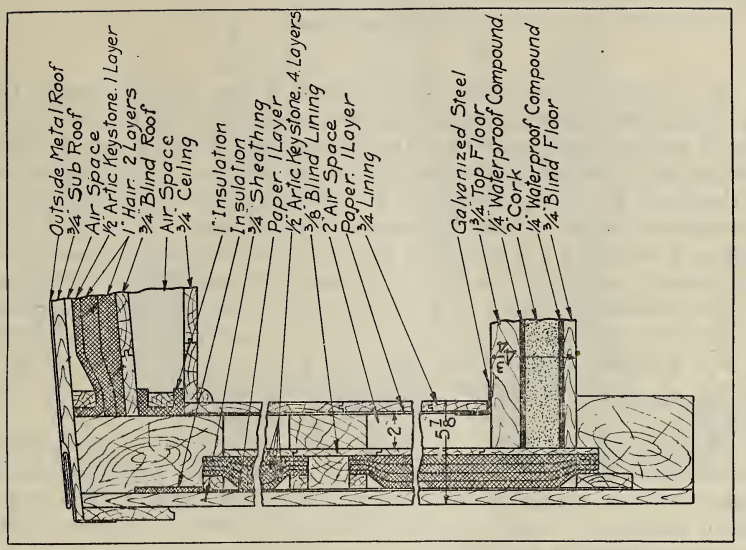

Fig. 15 Type of Refrigerator Car

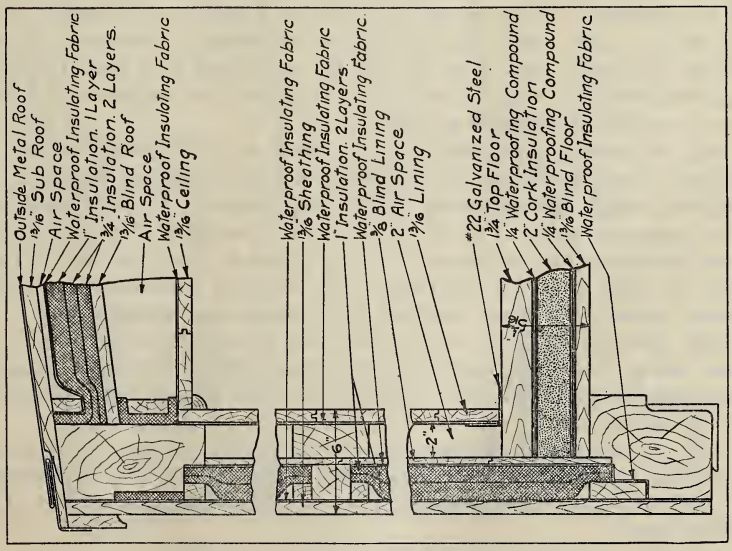

Fig. 14 Type of Refrigerator Car

Table 2 shows the thermal conductivity of white pine wood as 19.0. In this case C for white pine is

$$C = \frac{19}{24} = 0.790$$

97 Available information indicates that in the original formula $1/K_1$ was 2.0 but that modern practice indicates this to be too high a value. The value of this factor has been the subject of considerable discussion. A number of experiments indicate that surface resistances will vary and in some cases the value of $1/K_1$ will be as high as 0.8. However, this is under perfectly still air conditions and at ordinary atmospheric temperatures. It is felt that by assigning a value of 0.5 to $1/K_1$ that refrigerator-car conditions are very nearly approximated, because in this type of equipment there is some air circulation. For these reasons the value of $1/K_1$ is taken as 0.5 in the following calculations.

98 $1/K_2$ is another factor that has been the subject of much discussion, but it is generally conceded that its value is so small that it can be safely and for all practical purposes treated as zero and eliminated from the equation.

99 As previously indicated, the value of air space has been a subject of considerable discussion, because there is some question regarding the length of time that the so-called dead-air spaces remain tight and function as true dead-air containers. A great deal has been said in favor of filling air spaces with a light insulating material.

100 The following calculations have been made in two ways: first, by assigning an insulating value to the air space, and second, by eliminating it entirely. Comparative results are shown in Table 3. Laboratory tests show that the resistance of an air space is a direct function of its width up to about 0.6 in., after which the resistance is practically a constant. In the following calculations the value of D/C for air is used as 1 when the space varies from 0.6 in. to 6 in. in width. For example, the resistance of a $\frac{3}{8}$-in. air space would be 0.375/0.600 or 0.625.

101 In order to illustrate the application of the formula, as well as to indicate the difference in the efficiency of walls, roofs and floors in cars of different design, the following examples are offered. In the calculations the car cross-sectioned in Fig. 14 is used as an illustration of good construction and relatively high efficiency. The car shown in Fig. 23 is used in comparison in order to show the greater rate of heat transmission or lower efficiency caused by different methods of insulation and construction.

102 Some slight value can be attributed to such factors as waterproof papers, fabrics and compounds, but for other ordinary practical purposes these values are given but little consideration, and are generally eliminated from the calculation.

103 In considering the thermal conductivity in sheathing and lining, the thermal conductivity value of Virginia pine is used. Inquiry indicates that the conductivity of this material is practically the same as that of fir or yellow pine.

104 The calculations are made as follows.

105 *Roof.* — (Fig. 14). Using Equation (1) with corrected values from Table 2, the heat transmitted through the structure is calculated as follows:

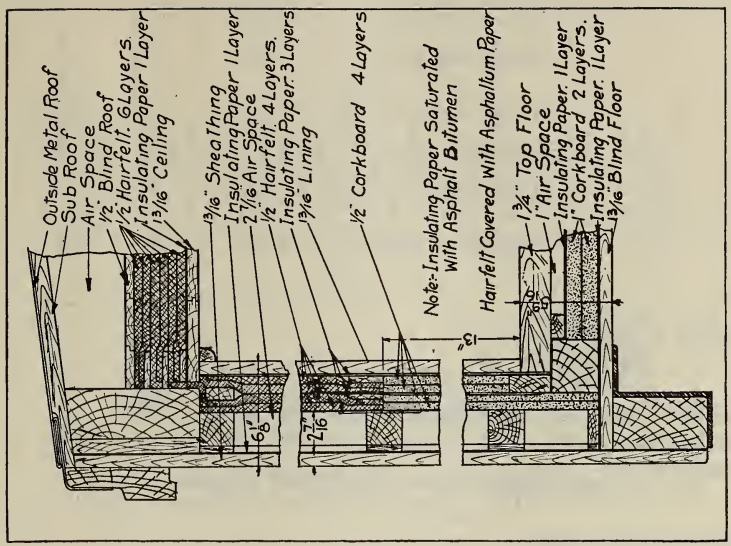

Fig. 17 Type of Refrigerator Car

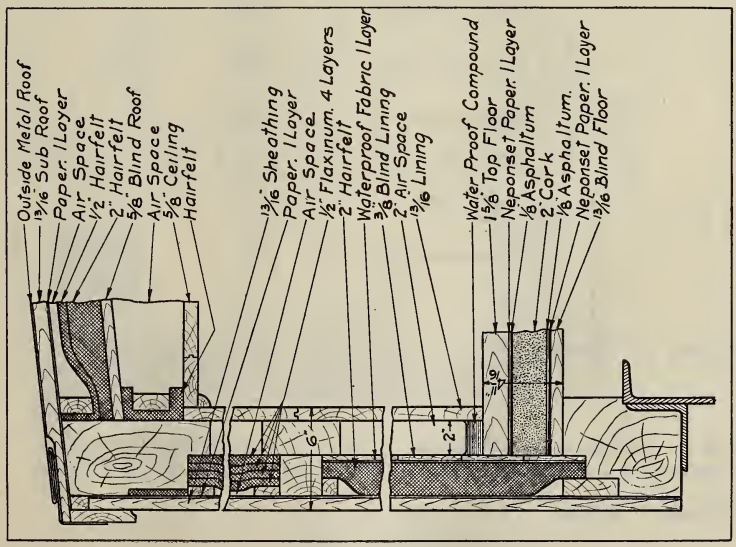

Fig. 16 Type of Refrigerator Car

$\frac{13}{16}$-in. sub-roof	$\frac{D}{C}$	= 0.847
$\frac{1}{2}$-in. air space		= 0.833
$2\frac{1}{2}$-in. hair felt	$\frac{D_1}{C_1}$	= 9.23
$\frac{13}{16}$-in. blind roof	$\frac{D_2}{C_2}$	= 0.847
$4\frac{1}{4}$-in. air space		= 1.0
$\frac{13}{16}$-in. ceiling	$\frac{D_3}{C_3}$	= 0.847
Surface resistance	$\frac{1}{K_1}$	= 0.5
	Total	= 14.104

Substituting in Equation (1)

$$H = \frac{1}{14.104} = 0.0709$$

Reduced to a daily basis (24 hours)

$0.0709 \times 24 = 1.702$ b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

Eliminating air spaces

$$H = \frac{1}{12.271} = 0.0814$$

Reducing to a daily basis

$0.0814 \times 24 = 1.953$ b.t.u. per sq. ft. per. deg. diff. fahr. per 24 hr.

106 *Walls.* — (Fig. 14). Following the same procedure we obtain

$\frac{13}{16}$-in. sheathing	$\frac{D}{C}$	= 0.847
2-in. hair felt	$\frac{D_1}{C_1}$	= 7.384
$\frac{3}{8}$-in. white pine	$\frac{D_2}{C_2}$	= 0.473
2-in. air space		= 1.0
$\frac{13}{16}$-in. lining	$\frac{D_3}{C_3}$	= 0.847
Surface resistance	$\frac{1}{K_1}$	= 0.5
	Total	= 11.05

Substituting in Equation (1)

$$H = \frac{1}{11.05} = 0.0905$$

Reduced to a daily basis (24 hours)

$0.0905 \times 24 = 2.172$ b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

Eliminating air spaces.

$$H = \frac{1}{10.05} = 0.0995$$

Reduced to a daily basis

$0.0995 \times 24 = 2.388$ b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

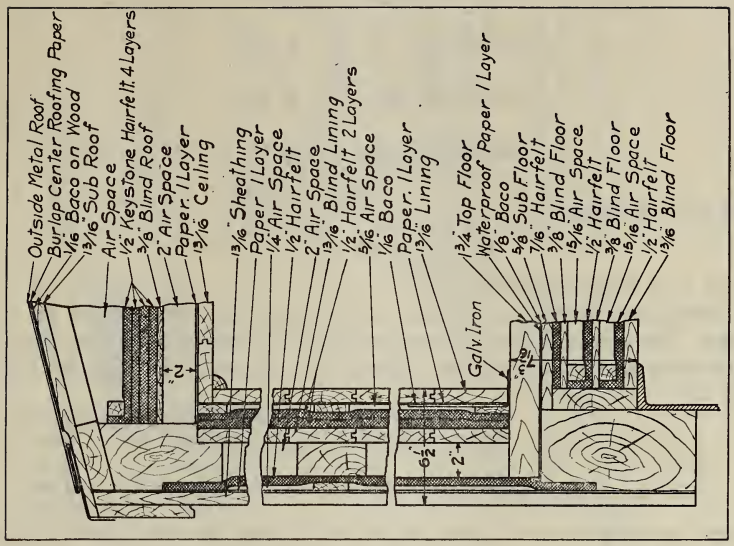

Fig. 19 Type of Refrigerator Car

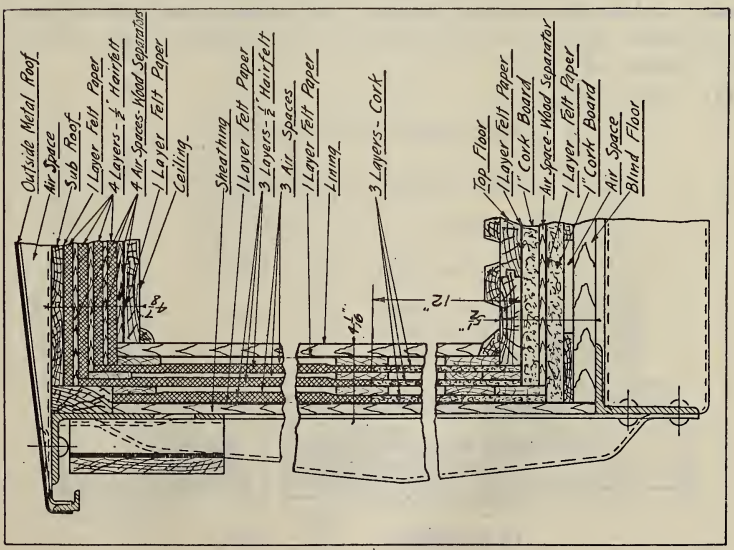

Fig. 18 Type of Refrigerator Car

107 *Floor.* — (Fig. 14).

$1\frac{3}{4}$-in. top floor	$\frac{D}{C}$	$= 1.826$
2-in. cork	$\frac{D_1}{C_1}$	$= 6.575$
$\frac{13}{16}$-in. blind floor	$\frac{D_2}{C_2}$	$= 0.847$
Surface resistance	$\frac{1}{K_1}$	$= 0.5$
	Total	$= 9.748$

Substituting in Equation (1)

$$H = \frac{1}{9.748} = 0.1026$$

Reduced to a daily basis (24 hours)

$0.1026 \times 24 = 2.46$ b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

108 The same methods followed with the car cross-sectioned in Fig. 23 give the following results which are stated in b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

	Including Air Space	Excluding Air Space
Roof	2.328	3.12
Wall	2.80	3.768
Floor	2.544	3.24

109 A comparison of these figures with those obtained from Fig. 14 is shown in Table 3, p. 41, and indicates and emphasizes the value of proper insulation and construction.

110 As the method of calculation in both cases is the same it is not necessary to show the complete details. The following roof calculations are shown, however, to indicate the treatment of the air spaces in cars with cross-section as shown in Fig. 23.

111 *Roof.* — (Fig. 23).

$\frac{13}{16}$-in. outside roof	$\frac{D}{C}$	$= 0.847$
$\frac{13}{16}$-in. sub-roof	$\frac{D_1}{C_1}$	$= 0.847$
Air space		$= 1.0$
$\frac{1}{2}$-in. hair felt	$\frac{D_2}{C_2}$	$= 1.846$
$\frac{3}{8}$-in. blind roof (white pine)	$\frac{D_3}{C_3}$	$= 0.473$
$\frac{3}{8}$-in. air space		$= 0.625$
$\frac{1}{2}$-in. hair felt	$\frac{D_4}{C_4}$	$= 1.846$
$\frac{3}{8}$-in. blind roof (white pine)	$\frac{D_5}{C_5}$	$= 0.473$
$\frac{5}{8}$-in. air space		$= 1.0$
$\frac{13}{16}$ in. ceiling	$\frac{D_6}{C_6}$	$= 0.847$
Surface resistance	$\frac{1}{K_1}$	$= 0.5$
	Total	$= 10.304$

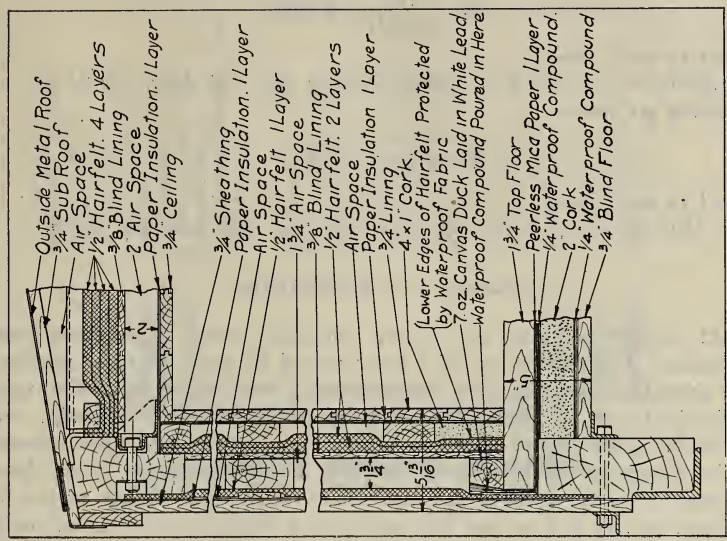

Fig. 21 Type of Refrigerator Car

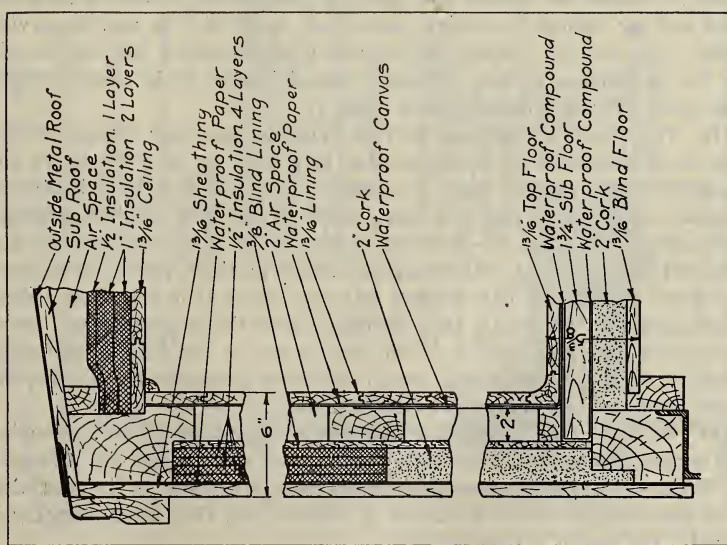

Fig. 20 Type of Refrigerator Car

Substituting in Equation (1)

$$H = \frac{1}{10.304} = 0.0970$$

Reduced to daily basis.

$0.0970 \times 24 = 2.328$ b.t.u. per sq. ft. per deg diff. fahr. per 24 hr.

Eliminating air spaces

$$H = \frac{1}{7.679} = .130$$

Reduced to daily basis

$0.130 \times 24 = 3.12$ b.t.u. per sq. ft. per deg. diff. fahr. per 24 hr.

MATERIALS AND WORKMANSHIP

112 Proper materials are a very important factor in refrigerator-car construction. The right grade of lumber should be used wherever required, and it should be properly dried before being placed in the car. Workmanship should be of the best. Insulation should be handled carefully, care being taken to see that it does not become torn or damaged. Such insulation placed in a car makes a weak link in a possibly otherwise strong chain. Some care in initial construction with attention to these details makes for an efficient car, as well as one that will have a longer life and lower maintenance cost than a car not receiving such attention.

113 While on the subject of materials, it is important to note the growing interest in the use of car lumber which has received preservative treatment. Lumber so treated has received considerable attention from car builders and car owners for several years, and much of it is now in service. Sufficient time has not elapsed to indicate what increased life can be obtained, but experience to date indicates treated lumber to be more durable, and one that will resist moisture and decay.

114 The Marsh Refrigerator Service Company has used creosote-treated lumber in certain parts of its refrigerator cars, such as sills, sub-floors and roof boards, and appears to be the pioneer in the use of treated lumber in refrigerator cars. The writer has been advised that this lumber is giving excellent service, and that no objection can be made to it on account of any odor caused by treatment. The treated lumber in these cars is submerged for a number of hours in hot creosote oil, after which it is placed in a drip-rack and permitted to drain. It is estimated that this treatment will result in large saving, doubling the life of the roofing boards and sills, and effecting considerable saving in labor that would otherwise be necessary to properly maintain these parts in the course of time.

115 An interesting report in connection with the use of treated lumber for use in the construction of cars was presented recently before the American Wood Preservers' Association. This report calls attention to the fact that decay is the principle cause of failures in lumber, and that great economy is possible by the use of a preservative.

116 It is evident that if some of the wooden parts of a refrigerator car can be made moisture proof or highly resistant to moisture, the efficiency of the car can be maintained at a much higher average.

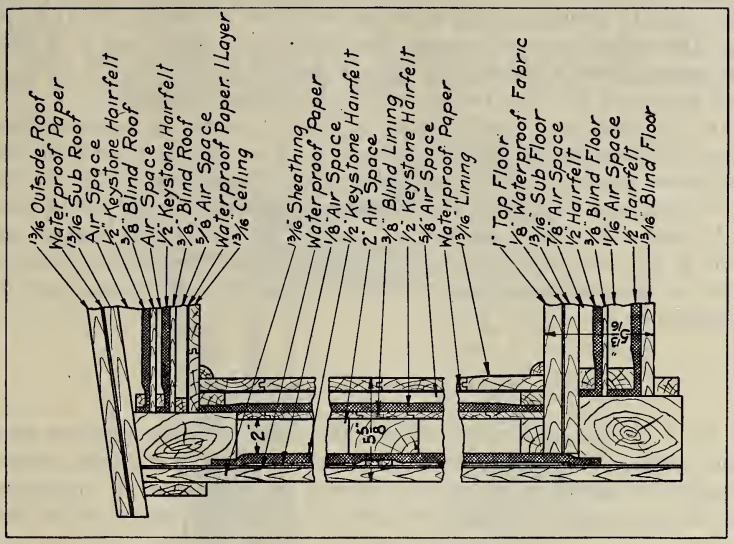

Fig. 23 Type of Refrigerator Car

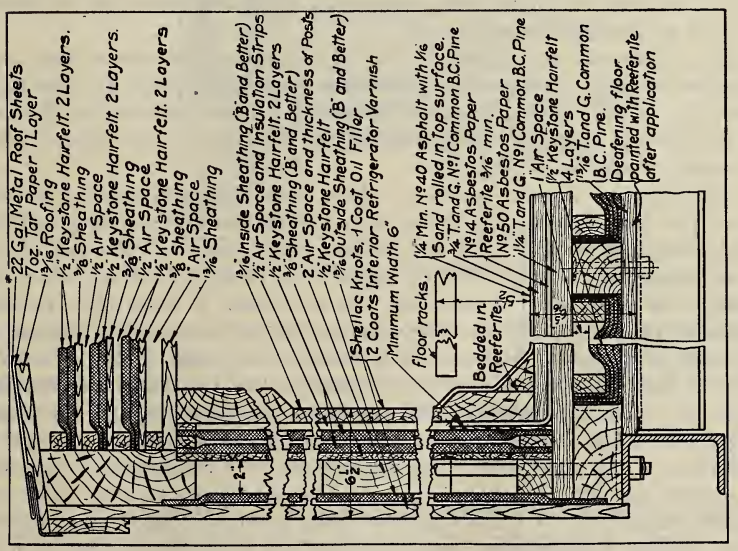

Fig. 22 Type of Refrigerator Car

117 The writer has been advised that some refrigerator cars are in service in which Balsa wood is the principal insulating material. This wood is very light in weight, having in its natural state a density of 7.1 lb. per cu. ft. It is a South American wood that grows very rapidly, and is of cellular structure. Table 2 shows it to have a thermal conductivity of 7.5 in its natural state and 8.3 when treated with waterproofing compounds.

118 It would be of great interest to know if treated or untreated Balsa wood is used between the ordinary walls of a car as insulation, or if the material figures largely in the construction of the superstructure of the car, such as lining and sheathing. Its strength is insufficient for its use in framing. It would also be of interest to know if the material is durable and efficient in this class of service, if any modification of car structure is necessary for its use, and if any reduction of car weight can be accomplished by its employment.

SOME OTHER SYSTEMS OF REFRIGERATION

119 A previous statement indicated that some reference would be made to other systems of refrigeration. In the cars described in the cross sections and tabulation, refrigeration is accomplished by means of air circulation, the air being cooled by contact with ice or ice containers placed at the ends of the car.

120 One modification of this system is a car in which ice containers are placed just below the roof and in the center of the car. In this system it is claimed that maximum refrigeration can be applied where the air within the car is at its highest temperature.

121 There do not appear to be a great number of cars of this type in modern service. The principal objections to such a system are decreased head room in the center of the car, weight of ice near the roof of the car, and difficulty of adopting this system for use with meat racks placed below the ceiling of the car.

122 Another system consists of a brine tank built into the roof at each end of the car. These tanks extend about 9 in. below the ceiling and are heavily insulated on top, sides and bottom. The tanks at each end of the car are connected to each other by pipes hung about 2 or 3 in. below the ceiling. The pipes are not insulated. In each tank is a partition running lengthwise of the car. In one partition are some check valves opening to the right; in the other partition some check valves open to the left. The theory is that when ice and salt are placed in the two tanks the swaying of the car in motion automatically. circulates the brine through the pipes, refrigeration being accomplished by contact of the air within the car against the surface of the pipes connecting the two tanks. Comparatively speaking, this system has not been in service a very great length of time. The advantages claimed for it are increased loading space, decreased consumption of ice, uniform temperatures, and a car that can easily be changed from a refrigerator to a heater car. The writer understands that these cars are being tested in various fields of service. It would be interesting to have some information regarding the ability of this system to supply refrigeration

when the car is not in motion, and what the system can accomplish in the way of quick pre-cooling when the car is placed at the loading shed or platform.

123 The interior of such a car is shown in Fig. 13. This illustration shows the floor racks propped up against the side walls so that the piping

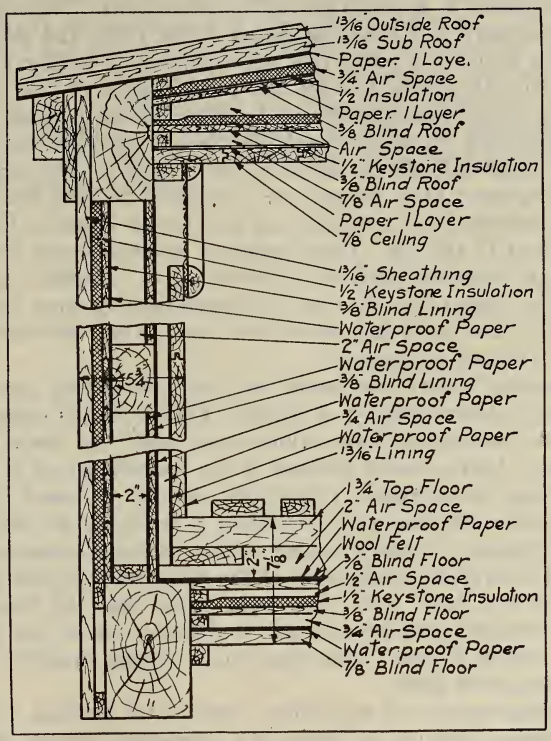

FIG. 24 TYPE OF REFRIGERATOR CAR

along the floor beneath the racks can be seen. This piping is used when the system is used to heat the contents of the car. Canvas troughs are placed beneath the piping located beneath the ceiling in order to catch any condensation or frost slush that may drop from these pipes.

PRE-COOLING

124 The importance of pre-cooling the lading and the resulting economy in the use of ice and labor were mentioned in a preceding paragraph. There are two distinct methods of pre-cooling cars and lading. The first is known as shippers' pre-cooling; and the lading is placed in cold storage rooms in which the proper temperature is maintained, and where the lading is allowed to remain until it cools to the proper temperature, after which it is loaded

quickly into cars that have been pre-iced. The second method is known as the carriers' pre-cooling, and generally consists of a system in which the car is loaded, after which the interior of the car and the lading are pre-cooled by mechanical means. In one system of this kind, at a point where a large tonnage of citrus fruit originates, cars are placed at a pre-cooling dock where the ventilator hatches are connected with large air pipes, and the lading is pre-cooled by means of a forced circulation of cold air. The air is brought to a low temperature in the cooling rooms of the plant, and passes through a long concrete tunnel and into a pipe connected to the hatchway at one end of the car. The air passes through the entire length of the car and out through another pipe connected to the hatchway at the opposite end of the car, and thence back to the cooling room through another concrete tunnel. The cold air passes through the car at a rate of 6,000 ft. per min. at a temperature ranging from 12 to 20 deg., for a period of 30 min., after which the pipes on the hatchways are reversed, and the cold air is forced through from the opposite end of the car. These pipes are reversed every 30 min. for a period of 4 hr. when the process of pre-cooling is complete. It is stated that citrus fruit loaded into the cars at temperatures of from 80 to 90 deg. can be brought down to 45 deg. within this time. After the car is cooled in this manner, it is iced to capacity.

125 Recently a very interesting and large pre-cooling plant has been erected in the Vegetable Section at Sanford, Florida, and is operated by the shippers. From this section large shipments of celery are made from January until June. After a careful analysis of the desirable points of an efficient pre-cooling plant, the shippers at this point decided to branch into an entirely new field by adopting a novel method of removing the heat from the celery. This consists of causing a large volume of cooled water to flow over the celery. The average temperature of the water used in this plant is approximately 35 deg. fahr., and it has been stated that the temperature of celery can be brought down from 75 deg. to within 2 deg. of the temperature of the water within 20 min. This plant has a cooling capacity of approximately 15 cars per 24 hours.

126 Another system of pre-cooling involves the freezing of the lading and a pre-cooling of the car by means of a brine-spray system. When commodities can be frozen hard and shipped in a pre-cooled car, refrigeration may be entirely accomplished or helped to a very great extent by the frozen lading.

127 Great economies are possible due to pre-cooling. Where small tonnage originates little pre-cooling has been done by mechanical methods on account of the high cost of the plant and equipment. Most mechanical pre-cooling is done where large tonnage originates. At such points the shippers frequently combine to built such a plant. Pre-cooling is receiving more and more attention in connection with various commodities and additional economy in the way of ice and labor may be expected.

GENERAL CONCLUSION

128 The inquiries upon which these few notes on railway refrigerator cars are based, indicate that a very great improvement has been made in

refrigerator-car construction and design, particularly within the last few years, but there is also evident indication that the field of investigation in connection with cars of this type is still a most fertile one. Some fairly recent cars indicate that subject of refrigeration in transit is not appreciated in some quarters as it should be. The subject of efficient refrigeration is a most important one, because cars that can be kept in continuous service with a minimum cost of maintenance and which are sufficiently efficient to protect the lading in transit, mean dollars and cents to the railways.

129 The writer apreciates that these notes are far from complete, but hopes that they may create some additional interest and help to re-emphasize the importance of the subject that has been ably emphasized in the past.

130 In conclusion the writer wishes to thank the railroads, car owners, car builders, manufacturers and individuals whose very generous assistance has made these notes possible.

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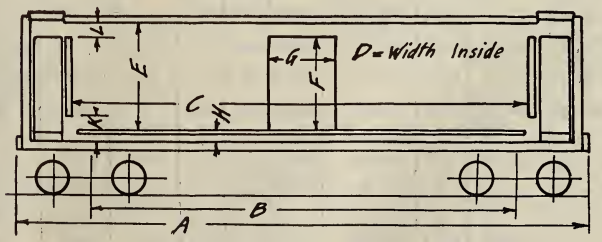

TABLE 1 DIAGRAM FOR REFRIGERATOR CARS

TABLE 1. REFRIGERATOR CARS

(See page 161 for key to dimensions)

Service	Road or Owner	A Over Striking Castings, ft-in.	B Truck Centers, ft-in.	C Between Bulkheads, ft-in.	D Width Inside, ft-in.	E Height Clear, ft-in.	F Door Height, ft-in.	G Door Width, ft-in.	H Height of Floor Rack	K Height Opening Below Bulkhead	L Height Opening Above Bulkhead	Type of Bulkhead	Tank or Bunker
Freight	1	42–5¼	31–2	34–4¾	8–7⅜	6–8 Meat Rack 7–3½ Ceiling	6–9⅝	5–0	5½″	7″	16½″	Solid, Swinging to Side	Brine Tank
	2	41–8½	30–8	33–2¾	8–2¾	7–0 3/16	6–8⅛	4–0	4 7/8″	12″	14″	Solid, Rigid	Basket Bunker
	3	42–1½	31–1½	33–2¾	8–2¾	7–0 7/16	6–8¼	4–0	4¾″	12″	12″	Solid, Rigid	Basket Bunker
	4	40–8½	29–10½	32–2⅜	8–0½	7–2	6–4¼	4–0	⅞″ Slat	.....	3½″	Swinging to Roof	Bunker (Collap.)
	5	34–0 Endsills	24–0	28–8	8–1	7–4½	6–0	4–0	....		17″	Solid, Swinging to sides	Brine Tank
	6	42–1½	31–1½	33–2¾	8–2¾	7–0 7/16	6–8¼	4–0	4¾″	2′–7″ 12″	14″	Solid, Rigid	Basket Bunker
	7	36–11⅛	26–0	29–10¾	8–2¼	7–1⅞	6–3	5–0	4½″	12½″	14⅞″	Solid, Swinging to side	Brine Tank
	8	37–9⅛ Endsills	26–10	36–0 Bet. Ends	8–3⅜	8–0 1/16	7–0	6–0	....	.....	.....	.....	Ice Boxes at roof
	9	37–10⅜	.....	31–0⅛	7–9⅜	7–1⅝	5–9¾	4–0	....	2′–6″	16¾″	Solid, Rigid	Brine Tank
	10	42–1½	31–1½	33–1¾	8–2¾	7–1	6–4¾	5–0	4¾″	15¼″	11½″	Solid, Rigid	Basket Bunker
	11	40–10 Endsills	30–9⅜	33–5⅜	8–2⅝	7–5⅛	6–4	4–0	....	16″	12″	Solid, Rigid	Box Bunker
	12	43–3½	32–0	34–6	8–10½	7–3½	6–3⅜	4–0	....	12″	12″	Solid, Rigid	Basket Bunker
	13	41–2	30–1	32–10½	8–10½	7–2 7/16	6–5	4–0	4¾″	12″	10 15/16″	Slat Syphon	Box Bunker
	14	40–9½ Endsills	31–9½	33–6¾	8–2¾	7–0 7/16	6–8¼	4–0	4¾″	.....	.....	Rigid	Basket Bunker
	15	40–0 Endsills	30–0	32–6⅞	8–3¼	7–4 13/16	.....	4–0	15/16″ Slats	11″	3¼″	Syphon, Swinging to Roof	Bunker (Collap.)
	16	40–0 Endsills	30–0	33–1½	8–3¼	7–4 13/16	.....	4–0	15/16″	15¾″	8⅛″	Solid, Rigid	Box Bunker
	17	41–8	30–10	33–2¾	8–2⅝	7–2½	5–9¼	4–0	....	12½″	12″	Solid, Rigid	Basket Bunker
	18	41–4¾ Endsills	31–6	33–0	8–4⅞	7–2⅛	5–8⅛	5–0	5″	.....	10″	Solid, Rigid	Basket Bunker
	19	40–0 Endsills	30–0	32–10	8–3⅝	.....	6–2⅜	4–0	....	.....	.....	Slat Syphon Rigid	Box Bunker
	20	41–10	30–2	32–7	8–4⅝	7–4	5–10	4–0	...	10″	8″	Slat Syphon, Rigid	Box Bunker
	21	38–7⅜	26–10½	30–11½	8–4	7–1	5–11½	4–0	5″	9″	14″	Solid, Rigid	Box Bunker
	22	41–11½ Endsills	31–8¼	32–10	8–3	7–1 3/16	6–3⅝	4–0	5″	11⅝″	12″	Solid, Rigid	Basket Bunker
	23	37–3⅛	26–4	30–4⅝	7–10⅞	7–2 9/16	6–2¼	4–0	....	2′–4″	21″	Solid, Rigid	Brine Tank
	24	42–6	32–6	32–11½	8–4⅜	7–3 3/16	6–2	4–2	Slat	12″	12½″	Solid, Rigid	Basket Bunker
Passenger	25	48–8¾	34–0	38–4¾	8–7⅜	7–3½	6–9⅝	5–0	5½″	7″	18½″	Solid, Swinging to sides	Brine Tank
	26	45–0 Endsills	33–8	38–10	8–1⅞	7–2½	6–3	5–0	4½″	10″	15″	Solid, Swinging to side	Brine Tank
	27	43–0	30–4	35–5	8–7⅜	7–4	6–3¾	5–0	4¾″	9⅜″	15¼″	Solid, Swinging to side	Brine Tank

TABLE 1. REFRIGERATOR CARS (*Continued*)

(See page 161 for key to dimensions)

Floors					Sides					Roof					Type of Underframe
Layers				Kind of Insulating Material	Layers				Kind of Insulating Material	Type of Outside Roof	Layers			Kind of Insulating Material	
Wood	Waterproof Paper or Fabric	Air Space	Insul. Mat'l.		Wood	Waterproof Paper or Fabric	Air Space	Insul. Mat'l.			Wood	Waterproof Paper or Fabric	Air Space	Insul. Mat'l.	
4	2 Paper	2	4	½" Keystone Hairfelt	3	.....	2	3	½" Keystone Hairfelt	Outside Metal	5	1 Paper	4	6	½" Keystone Hairfelt	Steel
2	1 Paper	..	1	2" Cork	3	2 Paper	3	3	½" Keystone Hairfelt	Outside Metal	3	1 Paper	2	4	½" Keystone Hairfelt	Steel
2	.....	..	1	2" Cork	3	3 Fabric	1	2	1" Insulation	Outside Metal	3	2	2	3	.....	Steel
3	1 Paper	1	4	½" Insulite	3	4 Paper	3	4	½" Flaxlinum	Wood	5	5 Paper	3	4	½" Flaxlinum	Steel
5	8 Paper	3	1	1" Hairfelt	4	6 Paper	2	1	1" Hairfelt	Wood	5	5 Paper	4	2	1" Hairfelt	Steel
2	.....	..	1	2" Cork	3	1 Fabric	1	2 1	½" Keystone 1" Hairfelt	Outside Metal	3	1 Fabric	2	2 2	½" Keystone ¾" Hairfelt	Steel
4	2 Paper	3	1 1	Keystone Woolfelt	3	4 Paper	2	1	Keystone	Wood	5	3 Paper	2	1 1	Keystone Sawdust	Steel
3	.....	4	3	½" Flaxlinum	3	2 Paper	3	2	½" Flaxlinum	Outside Metal	3	.....	2	4	½" Flaxlinum	Steel
5	6 Paper	3	1 1	½" Hairfelt 1" Hairfelt	3	3 Paper	3	1	½" Hairfelt 1" Hairfelt	Wood	6	.....	5	2	½" Hairfelt	Wood
2	.....	..	1	2" Cork	2	2 Paper	2	4	1" Flaxlinum	Outside Metal	3	1 Paper	2	1 1	½" Hairfelt 2" Hairfelt	Steel
5	1 Paper	2	3	½" Hairfelt	3	2 Paper	3	3	½" Hairfelt	Outside Metal	3	1 Paper	2	4	½" Hairfelt	Steel
5	4 Paper	3	4	Woolfelt	..	.....	..	..	.....	Wood	5	5 Paper	5	4	Woolfelt	Steel
4	1 Fabric	2	4	½" Flaxlinum	3	.....	1	4	½" Flaxlinum	Outside Metal	3	.....	2	5	½" Flaxlinum	Wood
2	.....	..	1	2" Cork	3	2 Paper	1	4	½" Arctic Keystone	Outside Metal	3	.....	2	1 2	½" Arctic Keystone	Steel
4	7 Paper	1	2	½" Flaxfelt	3	3 Paper	3	4	½" Flaxfelt	Wood	4	6 Paper	3	4	1" Hair ½" Flaxfelt	Steel Center Sills
4	1	2	4	½" Arctic Keystone	3	.....	3	4	½" Arctic Keystone	Wood	4	.....	3	4	½" Arctic Keystone	Steel Center Sills
2	.....	..	1 1	½" Insulite 1½" Cork	2	1 Paper	1	1 1	½" Insulite 1½" Insulation	Wood	3	1 Ruberoid 1 Waterproof Coat	1	1 2	2" Hairfelt ½" Insulite	Steel
3	.....	1	1	2" Cork	3	2 Paper	2	4	½" Insulation	Outside Metal	3	.....	2	4	.....	Steel
3	1 Paper	2	2	½" Linofelt	3	2 Paper	3	2	½" Linofelt	Wood	3	1 Paper	3	1 2	⅜" Insulation ½" Linofelt	Steel
5	1 Fabric	3	3	½" Hairfelt	3	4 Paper	3	2	½" Hairfelt	Wood	5	6 Paper	4	3	½" Hairfelt	Steel
4	1	2	2	½" Hairfelt	3	2 Paper	3	2	½" Hairfelt	Wood	5	2 Paper	3	2	½" Hairfelt	Steel
2	2 Paper	1	1	2" Cork	2	9 Paper	1	4 2	½" Hairfelt Cork at bottom	Outside Metal	3	13 Paper	1	6	½" Hairfelt	Steel
5	5 Paper	3	3	½" Hairfelt	5	5 Paper	4	3	½" Hairfelt	Wood	5	7 Paper	4	3	½" Hairfelt	Steel
2	3	2	2	1" Corkboard	2	2 Paper	3	3 3	½" Keystone Cork at bottom	Outside Metal	2	2 Paper	5	4	½" Keystone	Steel
3	2 Paper	1	4	½" Keystone	3	.....	2	3	½" Keystone	Outside Metal	5	1	4	6	½" Keystone	Steel
3	1 Paper	2	2	½" Keystone	4	1	1	4	½" Keystone	Wood, Canvas Covered	3	1 Canvas	2	4	½" Keystone	Steel
3	1 Paper	2	2	1¼" Tucork	4	.....	4	3	½" Keystone	Inside Metal	3	.....	3	4	½" Keystone	Steel

TABLE 2 THERMAL CONDUCTIVITY OF INSULATING MATERIALS

Material	Remarks	Thermal Conductivity [1]	Density [2]
Air	If no heat is transferred by radiation or convection	4.2	0.08
Calorax	Fluffy mineral powder	5.3	4.0
Kapok	Hollow vegetable fibers, loosely packed	5.7	0.88
Pure Wool		5.9	6.9
Pure Wool		5.9	6.3
Hair Felt	Fibers perpendicular to heat flow	5.9	17.0
Pure Wool		6.3	5.0
Slag Wool	Loosely packed	6.3	12.0
Keystone Hair	Hair felt and other fibers, confined with building paper.	6.5	19.0
Mineral Wool	Loosely packed	6.5	12.0
Corkboard	No artificial binder, low density	6.5	6.9
Mineral Wool	Fibers perpendicular to heat flow	6.9	18.0
Cotton Wool	Medium packed	7.0	5.0
Pure Wool	Very loose packing, probably air circulation through material	7.0	2.5
Insulite	Pressed wood pulp	7.1	12.0
Mineral Wool	Firmly packed	7.1	21.0
Linofelt	Vegetable fiber confined with paper, flexible and soft	7.2	11.3
Ground Cork	Less than $\frac{1}{16}$ in.	7.1	9.4
Corkboard	No artificial binder	7.3	9.9
Balsa Wood	Very light wood, across grain	7.5	7.1
Balsa Wood	Same sample with 13 per cent waterproofing compounds	8.3	8.0
Flaxlinum	Felted vegetable fibers	7.9	11.0
Fibrofelt	Felted vegetable fibers	7.9	11.0
Rock Cork	Mineral wool and binder	7.9	16.0
Balsa Wood	Across grain, untreated	8.3	7.4
Corkboard	With bituminous binder	8.4	16.0
Balsa Wood	Medium weight wood	9.2	8.8
Sawdust	Various	9.7	12.0
Air Cell (½-in.)	Corrugated asbestos paper, enclosing air spaces	11.0	8.8
Air Cell (1-in.)	Corrugated asbestos paper, enclosing air spaces	12.0	8.8
Asbestos Paper	Built up of thin layers	12.0	31.0
Balsa Wood	Heavy	14.0	20.0
Fire Felt Sheet	Asbestos sheet, coated with cement	14.0	26.0
Fire Felt Roll	Flexible Asbestos sheet	15.0	43.0
Cypress	Across grain	16.0	29.0
Asphalt Roofing	Felt saturated with asphalt	17.0	55.0
White Pine	Across grain	19.0	32.0
Mahogany	Across grain	22.0	34.0
Oak	Across grain	24.0	38.0
Maple	Across grain	27.0	44.0
Virginia Pine	Across grain	23.0	34.0

[1] Thermal Conductivity in B.t.u. per day (24 hr.) per sq. ft. per deg. fahr. per in. thickness.
[2] Density, lb. per cu. ft.

TABLE 3

Comparison of B.t.u. per sq. ft. per deg. diff. fahr. per 24 hr. in cars shown in Fig. 14 and Fig. 23.

	Including Air Space		Excluding Air Space	
	Fig. 14	Fig. 23	Fig. 14	Fig. 23
Roof	1.702	2.328	1.953	3.12
Wall	2.172	2.80	2.388	3.768
Floor	2.46	2.544	2.46	3.24

No. 1844

REDUCTION OF FUEL WASTES IN THE STEEL INDUSTRY

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Member of the Society

The paper sets forth the economy of utilizing blast-furnace gas, at one time considered to be of little material value, in reducing the wastes of the steel industry. It is shown how the quantity of gas to be expected in the production of one ton of pig iron may be computed, and the uses to which this gas may be put are analyzed quantitatively. A heat balance, based on the production of one ton of pig iron, is presented, showing the sources of heat and the requirements for blast-furnace and steel-mill operation. It shows that the total heat from the by-products of a ton of pig iron is more than sufficient to finish a ton of steel. The heating requirements are not met by the tar and by-product gas, although the average power requirements are more than met by the blast-furnace gas and coke breeze. It therefore becomes desirable to reduce the heating requirements for the production of steel by using more efficient heating furnaces, and to conserve all possible heat by charging hot steel into the soaking pit and reheating furnaces, in order to reduce the quantity of fuel required for supplementary heating.

In this period of readjustment there is every reason for the reduction or, if possible, the elimination of waste, and the managers and engineers of the steel and iron industry have been alive to the advantages resulting from the conservation of fuel. There are, however, in this and in all allied by-product industry, a number of opportunities for reduction in the consumption of fuel, some of which have been discussed many times and others only briefly touched upon.

2 In order to show the relative proportion of the energy in the shape of heat or power required for the manufacture of steel that is utilized from the fuel by-products of the manufacturing process, the following figures have been prepared showing the quantity of heat derived from the by-products of the manufacture of pig iron.

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ECONOMY OF UTILIZATION OF BLAST-FURNACE GAS

3 In the production of one ton of pig iron, from under 1800 to over 3000 lb. of coke are used, this variation being due to difference in metallic content of the charge, condition of the furnace lining, quality of coke, etc. The heat energy of this coke is utilized in the reduction of the ore to pig iron and changing the impurities to a slag; a part is lost in radiation, and approximately half is delivered at the top of the furnace in the shape of gas for use under boilers for the production of steam, or in gas engines for the production of power. The gas has also been used in soaking pits, in by-product oven heating, ladle drying, etc., and other uses have been proposed, but by far the greatest quantity has been used under boilers.

4 It has been only within the last few years that blast-furnace gas has been considered to have any material value. The value generally placed upon gas, and appearing as a credit to blast-furnace cost, has been based upon the cost of the coal replaced by the gas, with an allowance for the cost of firing coal; and even with comparatively inefficient boilers, and in many cases the lack of efficient supervision, the cost of steam in these boiler plants has been generally low. The coke consumption of the blast furnaces which largely determines the volume and B.t.u. value of gas was variable, and the steam consumption both of the blast furnaces and of other departments supplied from the furnace plant varied widely. The credit was largely an estimate, and at times when the gas supply exceeded the demand, steam or blast-furnace gas was wasted. The blast-furnace superintendent was generally interested mainly in obtaining as high stove heats as could be carried, regardless of how much gas was wasted at the stoves through leaving the gas on after they had become heated, or by forcing gas to the stoves above their efficient combustion capacity. Gas was similarly wasted at boilers, due to too small combustion chambers, inefficient gas burners, lack of proper baffling, or lack of proper attention paid to combustion. With nominal coal costs this waste might be excused but with the realization that coal deposits have a limit and with the increasing value of coal and its by-products, any waste of fuel should be stopped provided the cost of eliminating the waste is not excessive.

5 In the past there has been no recognized method of determining whether or not a blast-furnace plant has been guilty of wasteful practice, or if guilty, what the extent of this waste might

be, and it is the intention of this paper to propose a standard method of comparison of total combustion efficiency, both with the idea of determining, for any particular plant, the relative efficiency, from day to day or month to month, as well as for the comparison of one blast-furnace plant with another, for the information of the management and the engineers in determining the necessity for improvement in equipment or personnel. Reference is made particularly to the latter, for the reason that after twenty years' experience around blast-furnace plants the author has found that there is generally a greater gain in efficiency possible per dollar of expenditure by efficient attention to operation of existing equipment, than by making changes in equipment without proper supervision.

HEAT AVAILABLE FROM THE BLAST FURNACE

6 In proposing this standard comparison reference is made to Fig. 1 which is taken from Mr. Brassert's paper on Modern Blast Furnace Methods, published in the year book of the American Iron and Steel Institute in 1914. This chart shows the effect of variation in coke rate upon the heat value of blast-furnace gas at 62 deg. fahr. As blast-furnace gas leaves the top of the furnace it usually has a temperature of from under 300 to over 600 deg. fahr., generally averaging about 400 deg. fahr. At this latter temperature, the sensible heat of the gas is equivalent to about 8 B.t.u. per cubic foot of gas at 62 deg. The B.t.u. per cubic foot of blast-furnace gas according to Brassert is $62.5 + 0.016R$, where R = lb. of dry coke per ton of pig iron; and including sensible heat of the gas at 400 deg. fahr. would be equal to $70.5 + 0.016R$. The cubic feet of gas per ton of pig iron (as shown by Brassert) is equal to $71.4R$, so that in the top gas from a furnace at 400 deg. fahr. the B.t.u. per ton of iron is $1.142R^2 + 5033.7R$. Brassert also gives the cubic feet of air per pound of coke equal to $51.4R$, so that both gas production and the blast requirement bear a direct ratio to the coke consumption of the furnace per ton of iron. The figures given in Mr. Brassert's paper, so far as the author has been able to discover, are applicable to all blast furnaces using coke as fuel.

HEAT REQUIRED FOR THE STOVES

7 The blast is heated in stoves under pressure, and the heat required per cubic foot of blast is proportional to the rise in blast temperature in the stove, and inversely proportional to the efficiency

of the stove, which is usually considered to average 60 per cent. After the necessary gas is used to heat the blast, the remainder can be measured in the shape of power. Any increase in stove or boiler efficiency or reduction in leakage of gas, etc., is reflected in increased surplus power, and as we are largely interested in the surplus power, after the blast furnace requirements are satisfied, any error in the assumptions relative to the efficiency of the stoves is of minor importance, particularly as the stoves use the smaller fraction of the

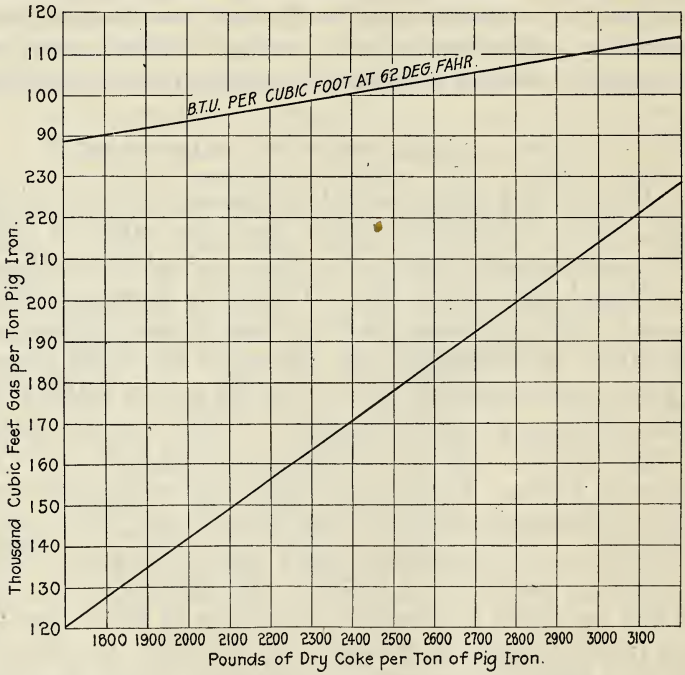

FIG. 1 EFFECT OF COKE RATE ON PRODUCTION OF BLAST-FURNACE GAS

total. The blast supplied from the blowing engines is heated by compression to about 150 deg. fahr., varying somewhat with blast pressure, etc., and is usually heated in the stoves to from 1150 to 1350 deg. fahr. In the following calculations it has been assumed that 0.02 B.t.u. are required per cubic foot of blast per deg. fahr., that the blast is heated in the stoves to 1250 deg. fahr., that the average over all stove efficiency (i.e. ratio of heat given to the blast to the heat in the gas) is 60 per cent and that 55 cubic feet of sblat are heated per pound of coke charged.

8 The heat required in the gas for heating the blast for one ton of pig iron to a temperature of 1250 deg. fahr. can therefore be expressed by the following formula where H = B.t.u. per lb. of coke burned in the blast furnace.

$$H = \frac{(1250 - 150) \times 0.02 \times 55R}{0.60} = 2020R$$

The heat in gas per ton product after the blast is heated is, therefore, $1.142R^2 + 3013.7R$. For other top temperatures and blast temperatures the constants will change slightly.

HEAT REQUIRED FOR GAS-FIRED BOILERS

9 In order not to enter into the discussion of the relative merits of steam vs. gas engine, surplus uncooled gas, after the stoves are heated, will be assumed to be used under boilers. In one boiler horsepower-hour there are 33,305 B.t.u., and assuming that the "standard" efficiency of blast-furnace-gas-fired boilers is 66.6 per cent, in one boiler horsepower-hour $\frac{33{,}305}{0.666} = 50{,}000$ B.t.u. would be required in the gas, so that the boiler hp-hr. per ton would be $\frac{\text{Surplus B.t.u. per ton iron}}{50{,}000 \text{ B.t.u.}}$. As the blast-furnace product is usually expressed in tons per day the average boiler horsepower per hour generated from blast-furnace gas under the above conditions can be expressed by the formula:

Average boiler horsepower per hour per ton of pig iron per day

$$= \frac{1.142R^2 + 3013.7R}{50{,}000 \times 24}$$

10 In assuming the standard efficiency of 66.6 per cent for boilers and 60 per cent for stoves, it should not be understood that these are the maximum efficiencies that can be attained, but rather that they are taken as a standard for comparison and represent average practice that can be maintained in a fairly modern plant correctly designed, and operated by reasonably intelligent men.

11 Using the above, Fig. 2 has been prepared. This chart also shows the per cent of top gas required for heating the blast to an average of 1250 deg. fahr. with the stoves of 60 per cent total

efficiency. This chart can be used by the blast furnace superintendent or engineer to determine the horsepower that should have been generated from gas for any period. From the actual results as shown by a meter on the boiler feed line, after deducting an allowance for steam generated by coal firing (usually small) or for

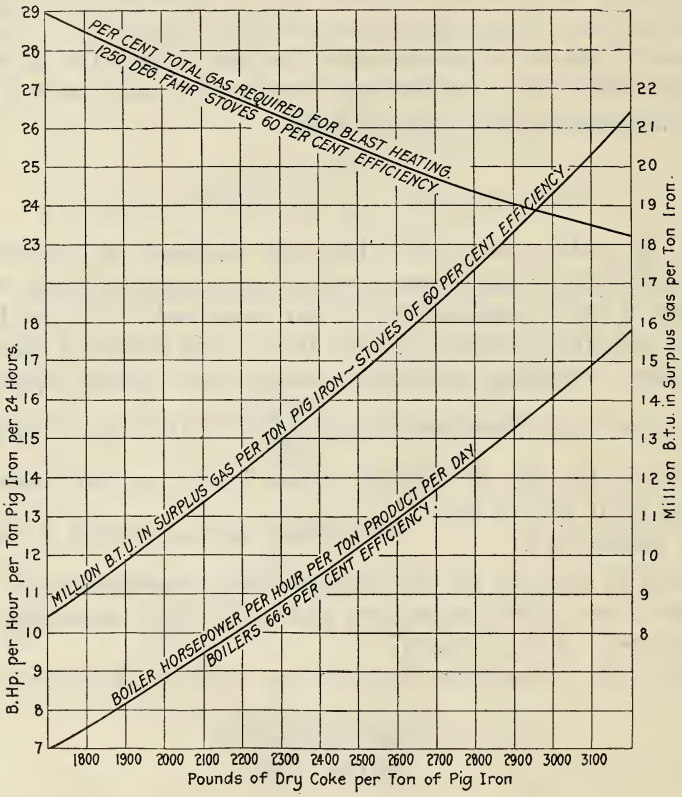

FIG. 2 EFFECT OF COKE RATE ON PRODUCTION OF POWER FROM SURPLUS BLAST-FURNACE GAS

gas used by gas engines, it will be possible to determine whether the total efficiency, including stoves and boilers, is above or below the standard.

12 The following table shows the boiler horsepower-hours per ton of pig iron for several coke rates as calculated from the above formula.

Pound of Coke per Ton	Boiler-Horsepower-Hours per Ton Product	Average Boiler-Horsepower per Ton per Day
2000	211	8.8
2200	242	10.1
2400	276	11.5
2600	310	12.9
2800	348	14.5
3000	386	16.1

13 To furnish the blast to the furnace with either high-pressure condensing, turbo-blowers or reciprocating steam engines requires about 70 boiler hp-hr. per ton of coke burned in the blast furnaces, and to operate the various furnace auxiliaries, such as pumps, air compressors, steam bells, mud guns, hoists, etc., requires from 15 to 25 boiler hp. per ton of product, so that after the furnace requirements have been met, there is available for the generation of power from 100 to 250 boiler hp-hr. per ton of pig iron, depending on the coke burned in the furnace.

14 At 2 kw. per boiler hp-hr., this means that from 200 to 500 kw-hr. per ton of pig iron is possible with steam equipment from the surplus gas from the blast furnaces.

HEAT REQUIRED FOR MANUFACTURE OF STEEL FROM PIG IRON

15 The manufacture of steel from pig iron in the open hearth requires heat for melting scrap and fluxing the impurities, and power for the various operations incident to this process. The quantities of heat and power required vary materially with different plants on account of difference in equipment, practice, etc., but for the purpose of this paper it will be assumed that a ton of ingots requires one ton of pig iron plus enough scrap to offset the losses and that this work requires from 5,000,000 to 9,000,000 B.t.u. and an expenditure of from 20 to 30 kw-hr. per ton. This ton of ingots will make about 0.85 tons of blooms, and the heating in soaking pits will require from 1,500,000 to 3,500,000 B.t.u. per ton of steel depending upon the temperature of the steel arriving at the pits, rate of operation, etc., and the rolling of the ingots into blooms will require from 40 to 60 kw-hr. per ton. From the blooming or slabbing mill the material, usually reheated at an expenditure of from 2,500,000 to 6,000,000 B.t.u. per ton, is furnished to finishing

mills, the power required, depending on the reduction of section, temperature of rolling, etc., varying from 35 to over 300 kw-hr. per ton.

UTILIZATION OF THE INDUSTRIAL WASTES

16 Waste heat from open hearth and heating furnaces and kilns is utilized for the production of steam, and otherwise waste steam is utilized in low or mixed pressure turbines for the production of power, but as these subjects have been discussed many times, and are not peculiar to the steel industry, they will not be discussed except to say that while these economies are many times worth while in existing plants, very few new installations as designed include waste-heat boilers with furnaces, or low-pressure turbines as integral parts of new power plants.

17 Surplus by-product gas, after the ovens are heated, is sold for use as domestic fuel, etc., replacing natural gas, water gas, etc., and is used in the steel plant for melting steel in the open hearth, and for various heating operations in the steel industry. In former years this material was considered as having only nominal value and only rarely was any attempt made to utilize it efficiently.

18 With the advance in fuel prices, shortage of natural gas, etc., and also largely due to special heating requirements connected with the manufacture of artillery and ammunition during the war, a number of combustion systems were developed that by more or less accurately regulating the proportion of air to gas have greatly increased the combustion efficiency of heating furnaces, and the control of combustion has been found of great importance in the maintenance of economy.

19 Dust coke breeze, from the coke plant (i.e. the material through a $\frac{3}{8}$-in. screen), is used to cover molten pig iron in transit from the furnaces and in making bottoms of soaking pits and heating furnaces. A great deal of this material which was thrown away or more or less inefficiently burned on hand-fired grates or, by mixing with slack, on mechanical stokers, is now more efficiently burned by itself, on traveling-grate forced-blast stokers.

20 Tar, either before or after the creosote oils have been removed, has been found to be a very good fuel for open-hearth furnaces, either by itself or in conjunction with by-product gas, and has been used for steam production either as an auxiliary fuel on blast-furnace boilers or by itself, and its use has been proposed to heat by-product ovens, thus releasing additional gas otherwise required for heating the ovens.

HEAT BALANCE

21 Table 1 gives an example of the approximate heat balance of one ton of pig iron as used in the steel plant, using average figures, and based on an average coke consumption in the blast furnace of 2500 lb. of coke per ton of product.

TABLE 1 HEAT BALANCE FOR MANUFACTURE OF ONE TON OF PIG IRON

From the coke plant delivering one ton of coke there result the following fuel by-products for consumption in the steel plant after the heat and power requirements of the by-product plant have been met:

	B.t.u.	B.t.u.
12 gal. tar at 162,500 B.t.u. per gal., B.t.u.	1,950,000	
30 lb. of coke breeze at 12,000 B.t.u. per lb., B.t.u.	360,000	
10,000 cu. ft. gas at 500 B.t.u. per cu. ft. (net value), B.t.u.	5,000,000	
Total per net ton coke		7,310,000
Heat available per ton pig iron		
In tar from by-product plant	2,435,000	
In coke breeze from by-product plant	450,000	
In surplus gas	6,250,000	
Surplus from by-product plant		9,135,000
In blast furnace gas		19,635,000
Required for blast heating	5,000,000	
Required for blowing equipment	4,400,000	
Required for blast-furnace operation	1,000,000	
Required for total blast furnaces		10,400,000
Surplus from blast-furnace plant		9,235,000
Total heat energy from by-products		18,370,000
Heat required for steel plant		
Open hearth (pig iron to ingots)		
Melting (approx.)	7,000,000	
Power, 25 kw. at 25,000 B.t.u. per kw-hr.	625,000	
Total heat consumption		7,625,000
Roughing Mill		
Heating (approx.)	2,000,000	
Power, 50 kw-hr.	1,250,000	
Total		3,250,000
Finishing Mill		
Heating (approx.)	4,000,000	
Power, 100 kw-hr.	2,500,000	
Total for finishing		6,500,000
Total heating requirements	13,000,000	
Total power requirements	4,375,000	
Total		17,375,000

NECESSITY OF HEAT CONSERVATION

22 From Table 1 it will be seen that the total heat from the by-products of a ton of pig iron is more than sufficient to finish the average ton of steel, but upon inspection of the table it will be

found that the heating requirements have not been met from tar and by-product gas, while the average power requirements are more than met from blast-furnace gas and coke breeze.

23 It therefore becomes very desirable to reduce the heating requirements for production of steel by using more efficient heating furnaces, and to conserve all possible heat by charging hot steel into the soaking pit and reheating furnaces, in order to reduce the quantity of fuel required for supplementary heating.

24 The production of gas at the by-product plant is practically uniform, while the consumption at the steel plant is variable and as a result the gas pressure at the steel plant (usually some distance away from the by-product plant) is variable. As gas holders of sufficient capacity to absorb the fluctuations have not been found advisable it becomes necessary to schedule the use of gas by consuming departments in order partially to equalize the consumption. In many cases it is advisable to use by-product gas under boilers (although it is more valuable for other purposes) in order to prevent or reduce its waste, as it is easier to change from gas to coal on boilers than to fire up producers.

25 One of the most important problems of the steel plant manager and engineer is so to regulate the production and consumption of the various fuels occurring as a by-product of steel plant manufacture that the consumption of raw coal in producers or under boilers that would otherwise be sent to the by-product plant is a minimum, for the reason that the by-products from the coal, coked in the by-product plant, usually exceed any possible return as fuel.

DISCUSSION

F. J. Crolius. The author's effort is a fine contribution and mere non-constructive criticism would be unfair and of no value, — but he does not go far enough. His suggestion of a *method* of balancing elements, so that the smallest amount of *added* fuels will be consumed, is excellent. There is every probability that sufficient power may be derived as a by-product from the fuels necessary to the metallurgical operations to convert the original ore into finished steel.

But at present there is no steel plant designed, built and operated with that end in view. Fuels have always been an inci-

dent in productions, and any conclusions based upon present practices would be of small value. What we accept as "averages" today will be obselete within a few years.

In a recent article, the writer developed a graph which visualizes this conversion and its power resultants. No absolute figures were used, nor can be used, as there are, in all the calculations, enormous variants which include every detail of the complicated processes involved in the manufacture of steel. When we think in terms of low fuel and power costs, we seldom realize, for instance, that a limitation on mill out-put caused by a shortage of shear capacity may be a tremendous factor, and when we speak of ore-bin operation, we seldom realize that the doors of a set of ore bins operated by old steam cylinders blowing continuously, due to leaks, may require more than 1000 hp. as against 15 hp. if operated by an electric motor, which consumes power only when in service. The mere location of ingot strippers, or the inadequacy of a badly organized internal transportation system, including faulty crane service, is of vital importance in the fuel problem, although casual observation would scarcely include those items.

These details have been specified merely to draw concrete attention to the great difficulty of arriving at a common base for valuable comparisons. To visualize the possible mass savings, the writer assumed for an economic unit an iron and steel development which comprised a coke-oven capacity of, say, 12,500 tons of coal charged daily, sufficient blast-furnace capacity to utilize this coke in a given iron reduction, sufficient open-hearth capacity to absorb the iron product, with the normal charges of hot-metal and average scrap, the total ingot tonnage rolled to average sizes of blooms, billets, slabs, plates and structural. The figures are graphically represented in Fig. 3.

All coke-dust from the by-product plant and blast-furnace screens was burned under boilers, all by-product gas was burned in heating furnaces or soaking pits, all tar was burned in the open-hearth; steam was generated from blast-furnace gas, and all waste-heat gases were recovered by means of steam boilers or open-hearth and, possibly, heating furnaces.

There is a deficit in the heat-balance, using average figures of best present-day practice; but it is quite a fair assumption that an ideal plant, designed to conserve heat and power at every point, could be operated without the necessity of firing additional fuel.

The author's suggestion of better designed heating furnaces, while excellent as a suggestion for projected plants, is more difficult of accomplishment, and hardly justified in the matter of investment, than the many possible improvements or betterments which may be made and are being made to existing furnaces.

There is no question that heating furnaces, as usually operated, are violations of combustion laws, but after all, the losses at this point are small in comparison with those at other points.

Much can be done toward improving burner equipment, great savings are possible by adequate wall and cover construction and sealing, leaking valves are a source of constant loss, mere lubrication of valves will be found to return unusual dividends, dampers are for the most part uncontrolled, a mere incident in usual operation, almost no metering and recording devices are installed, without which no adequate comparisons can be expected.

What holds true of heating furnaces and soaking pits is equally true, and of greater importance in open-hearths, blast-furnace stoves and boilers.

Progress in steam generation efficiencies started with the automatic control of dampers and fuel and air-supplying equipment, and with recording devices which reliably pictured the daily operation. Pressure and draft gages, flow controls, pressure regulators, gas-analyzing instruments, thermometers and pyrometers are now available, and these used by an organized experimental crew will work marvels in any average steel unit and without which no steel unit can compete.

The author stresses organization and management and all will agree that herein lies the starting point for reduction of fuel losses in the steel industry, — and not only fuel losses but those myriads of small losses incident to any operation where fuel losses are tolerated.

R. S. Sage[1] and E. Pragst.[1] In this very instructive paper, the author shows us that for an average steel plant using 2500 lb. of coke per ton of pig iron produced, and allowing 175 kw-hr. per ton of pig iron, there is an excess of by-product fuel, satisfactory for use for power generation but unsatisfactory for process heating. This excess exists even after a most liberal allowance for blowing and rolling operations has been made. Accepting these figures as

[1] Power and Mining Engineering Department, General Electric Co., Schenectady, N. Y.

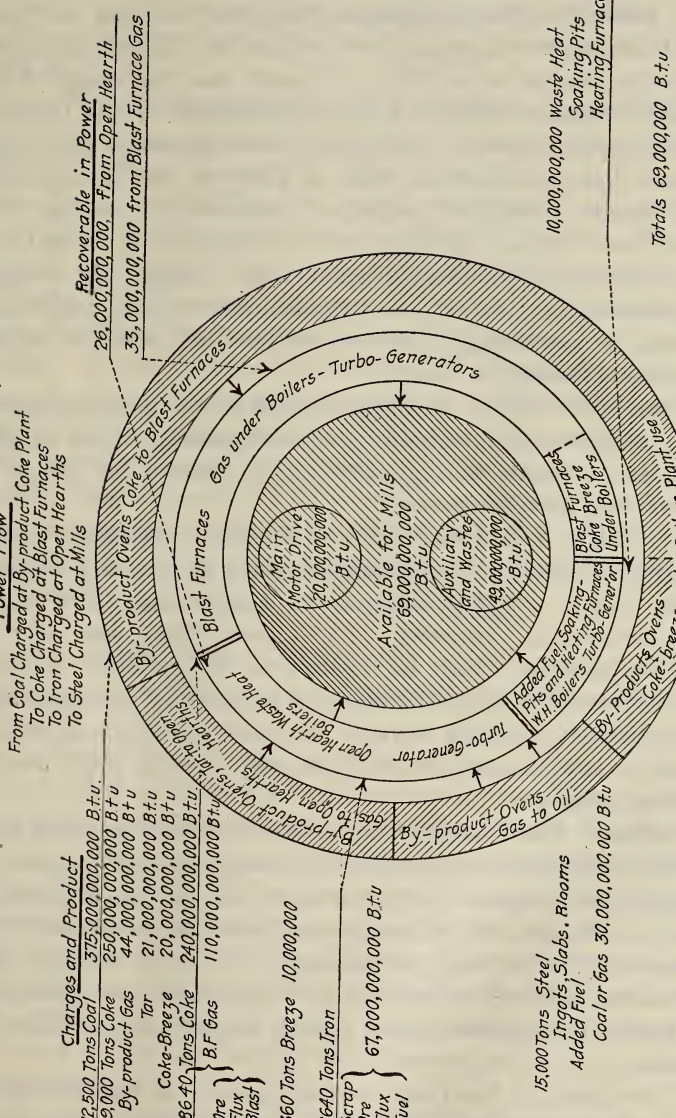

Fig. 3 Diagram of Power Flow from Coke Charged at By-Product Coke Plant to Coke, Iron and Steel in Furnaces and Mills

correct for the representative case which has been chosen, one must conclude in general that we who are primarily interested in the power and blowing plants of the steel industry are confronted with two entirely different classes of problems. On the one hand we have those plants where an excess of gas, etc., above power and blowing requirements, is available. In such cases the design of the power and blowing plants need not be influenced by the question of plant thermal economy. The logical development in such cases is the least expensive plant in terms of first cost and maintenance that will assure a sufficient degree of continuity of service. On the other hand, we have plants, some working with a much lower coke consumption than that assumed, some rolling cold sections and to small sizes, and others operating under both conditions, where there is a shortage of by-product fuel or would be unless high thermal economy is sought.

In view of the title of the paper, we feel that it might be appropriate to mention briefly some of the advancements made in generating-station design, as they have direct bearing on plants of the second just-mentioned class.

In seeking to reduce fuel waste in the steel industry, we must not lose sight of the fact that most products of the steel industry are being sold in a highly competitive market and, as a consequence, possibly unfortunately, fuel conservation can only be carried to a point at which the investment in apparatus required and cost of operating it for effecting the saving does not materially offset the monetary saving in fuel. In other words, we must produce steel at a minimum cost and secondly save as much fuel as possible in doing so.

Avoiding a direct comparison between the relative total costs of blowing and generating electrical energy through the use of gas engines as compared with turbines and confining ourselves primarily to the gains in economies now possible in steam-electric and steam-turbine blowing installations, we might mention that there has been a steady increase in the possible thermal efficiencies which might be expected from plants employing this class of equipment.

Up to recently it has been common practice in the steel industry to maintain two sets of boilers; one operating with blast-furnace gas, the other equipped with stokers and used in augmenting the output from the gas-fired boilers during periods of gas

deficiency. Recent developments in powdered-coal firing substantiated by operating experience make it now possible to use but one set of boilers in which provision is made to augment gas with powdered coal, thereby reducing the number of boilers required, bettering their operating load factor and increasing their thermal efficiency.

A further considerable saving in fuel is made possible by taking advantage of the full temperature range which is now possible in turbine operation. With reasonably cool condensing water it is now possible to maintain and effectively use throughout the year vacua that will average close to 29 in. The upper range of the heat cycle has been extended until now in large installations high-pressure superheated steam at temperatures as high as 750 deg. fahr. can be successfully employed for power generation. In the smaller plants, where it is desirable to seek high thermal economies, a steam temperature of 650 deg. fahr. can be conservatively recommended. Adding to the reduction in fuel made possible through the extension of the heat cycle that resulting from improvements in turbine thermal efficiency as a result of the advancement in the art of turbine design, it will be seen that a considerable saving of our fuel resources is now possible in a new installation over that of only a short time ago.

F. L. Estep. This paper is on a very important subject and emphasizes some phases of it which should be of considerable help in effecting fuel economies in the steel industry. The conditions are far from ideal in most steel plants and in some of them are still so bad that the writer feels that what the author describes as the inefficiency of former years is in many plants the inefficiency of today; and it is one of the outstanding features of this entire subject that there is generally much more to be gained by proper operation of existing equipment than by the installation of additional equipment. This is proved by the fact that in one steel plant within the last five years the coal consumption was reduced to less than half simply by the proper combustion of the blast-furnace gas, and by making an improvement in the economy of the steam users.

In very recent years blast-furnace boilers have been built with the firebox entirely too small and with insufficient draft. It was only within the last two years that one of the prominent

boiler manufacturers in this country started to make a difference in size of damper area between their boilers for coal-firing and their boilers for blast-furnace gas.

Although the following figures are given by the author as general average results the writer would like to criticise some of them, as they seem to be somewhat out of line with blast-furnace practice in general throughout the country, particularly in the North. For instance, 60 per cent stove efficiency seems rather low in a plant where there is any intelligent supervision, — 65 per cent would have been a better assumption. This statement is based upon a hot-blast temperature of 900 to 1000 deg. fahr. A hot-blast temperature of 1250 deg. is an exceedingly high assumption, as the above figures more nearly represent the general average practice. The assumption of 55 cu. ft. of air per lb. of coke leaves very little for leakage. This leakage has been found by actual tests as high as 50 per cent in some instances and 10 to 20 per cent represents a good average for most blast furnaces. Air leakage in blast furnaces where turbo-blowers are used is a much more simple proposition than with the use of blowing engines, as the question of leakage in the engine is eliminated and this has been found in some cases to be a large proportion of the total leakage.

In the table of horsepower the figures for total gas and total boiler horsepower seem higher than in ordinary practice. An allowance for leakage at the furnace top may have been omitted. Allowance should be made for this leakage as there is a loss at the furnace top which cannot very well be avoided.

The use of a table such as the author suggests for determining the expected boiler horsepower is a very great help to the operating force. It has been tried during 1919 at one northern steel company with figures similar to those given and the *expected* was checked by the *actual* horsepower daily for several months. As long as the blast furnace was running well the horsepower realized was between 93 per cent and 105 per cent of what had been figured, but as soon as the blast furnace began to work badly, the boiler horsepower fell off very rapidly and averaged only between 60 per cent and 80 per cent. This is probably due to the difficulty in burning the bad gas and to the increased per cent required by the stoves.

The writer believes that net tons of coke consumed is a better

basis than tons of iron produced for figuring the expected horsepower. The horsepower is more nearly in proportion to the coke used than to the iron made, and also is a direct result of the use of the coke. The boiler horsepower per net ton of coke would vary only about 20 per cent between 2000 and 3000 lb. of coke consumed per ton of pig, where the horsepower per ton of iron increases almost 100 per cent.

Table No. 1, together with the assumptions made in Par. 15, would lead the writer to think that no allowance has been made for the power required by auxiliary equipment. The most accurate figures the writer has for a very carefully studied steel plant with all the mills electrically driven show that the power required for driving the mill rolls plus the power for the blast furnace is only about 60 per cent of the total power necessary for the complete plant. The remaining 40 per cent consists of power used at the coke plant, general water-works power and all miscellaneous auxiliary power. In the latter item is included power for the calcining plant and all mill auxiliaries, such as cranes, tables, shears, finishing machinery, hydraulic pumps, etc., and also the necessary heat to keep the plant in condition during break-downs, delays, and Sunday shutdowns. If this 40 per cent is added to the author's table it shows a deficit of between five and ten million heat units, instead of a surplus of about one million.

This subject has been discussed many times before but it has not been discussed often enough nor thoroughly enough. In many plants, the equipment consists of good apparatus not properly installed, in some cases, and not properly kept up in others, and in a great many without proper thought and consideration being given to the importance of the efficiency of the various units.

The writer's attention was called some months ago to a case where an investigation by a competent disinterested party showed conclusively that this plant was burning one million dollars' worth of coal more per year than would be necessary with very slight modifications and changes in their equipment and with better supervision. It remains a fact, however, that up to date, this company has not made any changes.

Simplicity of apparatus so that the operating forces can keep it in efficient shape, should always be considered in preference to the need of a complicated system effecting fuel economies, which adds so much to the work of the steam department of the mill that they are liable to lose interest in the entire situation.

Walter N. Flanagan. The author's paper is very pertinent for the steel-works engineer or executive as he dwells first upon the use of blast-furnace gas. This is the keystone to power economy in the blast-furnace and steel plant as it is far easier to obtain and to maintain high efficiencies when burning blast-furnace gas than when burning coal. This is due not only to the fact that closer approximation to perfect combustion can be obtained with gaseous fuels, but also to the fact that in spite of the fluctuations in gas supply the load is more constant than the demand for coal firing in a steel works.

Blast-furnace gas furnishes an admirable opportunity for the application of air heaters for preheating combustion air.

The easiest and cheapest way for the average plant to obtain good economy in the use of blast-furnace gas is by the installation of proper burners. These need not be complicated, since for the usual installation of a large number of small boilers a very simple type of aspirating burner can be installed. This type of burner will, within wide limits of gas supply, automatically maintain very nearly the proper proportion of air for combustion. It requires practically no attention or repairs. Furthermore, the fact that the boilers are old or replacement is contemplated need not prevent immediate economy as the cost of installation will be returned in savings five or six times per year in most cases.

The saving by good burners is not only due to more perfect combustion, but also to quicker combustion. Both result in a lower stack temperature as secondary combustion among the tubes is eliminated and more heat is absorbed by radiation from the hotter fire.

A good gas cleaner is another source of saving with blast-furnace gas. Dry-cleaning methods offer great possibilities of saving, not only fuel, but also labor and flue dust. The sensible heat in 400-deg. gas mentioned in the author's paper is worth $22,000.00 per furnace per year in equivalent coal, in addition to the higher efficiency obtained by higher flame temperature and higher heat absorption by radiation. However, unless the gas is clean this value is lost in labor cleaning stoves, boilers and gas mains, in loss of flue dust and in decreased stove capacity. Flue dust once regarded as waste is now being utilized.

With dry-cleaned gas, the cleaning of gas mains or burners is eliminated, cleaning of boilers and stoves is reduced to a mini-

mum and the way opened for the obtaining of the maximum economy in the use of furnace gas. The importance of the proper preparation of the gas is indicated by the fact that the sensible heat saving is less than one-third of the total saving possible by dry cleaning.

Coke breeze screened out at the blast furnaces is being used successfully in many plants not equipped with the ideal means of burning it, namely, traveling grate stokers. When burned on underfeed stokers, either straight or mixed with coal, coke breeze will produce slightly more than one-half as much steam per pound as good coal. This is due to lower heating value, higher ash content, which means higher ash loss, and lower absorption of heat by radiation, which causes higher stack temperature. Its actual value on a money basis is somewhat lower as repair costs of stokers and fuel-handling machinery are higher.

W. Trinks. The paper shows that, with reasonable care, the fuel which goes to the blast furnace, and the by-products from the fuel are sufficient to operate a steel plant of the type which the author considers. The danger is that the conclusions which are drawn from this paper may be generalized and that readers may gain the impression that an economical blast furnace and steel plant can get along without coal, and still use steam-blowing equipment.

In northern blast furnaces, the coke consumption is more likely to be 1800 lb. per ton of pig iron, than 2500 lb. A few furnaces get along with 1700 lb. The effect of this reduction of coke consumption is twofold. First, the gas quantity is reduced, and second, its heating value is decreased. The final result is that the available combustion per hour is reduced in the ratio of

$$\frac{1.142 \times 1800^2 + 5033.7 \times 1800}{1.142 \times 2500^2 + 5033.7 \times 2500}$$

which roughly equals 0.65. This means that only 12,700,000 B.t.u. are available per ton of pig iron instead of 19,600,000 in the blast-furnace gas. Reduction in the coke quantity means a corresponding reduction in the quantity of by-products such as coke breeze, tar, and coke-oven gas.

Hence, $\frac{1.8}{2.5} \times 9{,}130{,}000$ B.t.u. equals 6,600,000 B.t.u. which are

available per ton of pig iron. With these figures, the total surplus heat energy from blast-furnace gas and from by-products is reduced to 9,000,000 B.t.u. per ton of pig iron, which is less than one-half of the amount given by the author for a southern plant.

In northern plants hot pig iron and scrap are used in approximately equal quantities, so that one ton of pig iron will certainly produce at least 1.5 tons of steel. This means that the heat requirements of the steel plant, including the open hearth, the blooming mill, and the finishing mill, must be multiplied by 1.5, which results in a heat consumption of 1.5×17.4 millions or 26.2 millions. But the surplus from the blast furnace and the coke plants equals 9 millions, and we have a deficiency of over 17 millions B.t.u. per ton of pig iron, which must be met by the burning of coal.

It is admitted that the open-hearth and heating furnace practice in a plant such as the author considers, is quite wasteful and that much better results can be obtained with good engineering, but even with better practice the deficiency cannot be wiped out.

As matters stand today, the writer knows of no low-coke-ratio plant with blooming mills and finishing mills in which the blast-furnace and coke-plant fuel suffices for the whole plant, unless gas blowing engines are used at the blast furnaces.

The Author. Mr. Crolius apparently overlooked the intention of this method of comparison of blast furnaces. He goes into some detail in showing that one plant will vary from another, and how this difference will be caused, depending on the type of equipment used.

This is one view of the comparison, it is true, but there is no reason why the same company cannot be compared day by day, the method suggested in the paper.

He also stated that he did not believe it would be advisable to improve existing heating furnaces. In that respect I beg to differ very strongly from Mr. Crolius, as we have found it is very advisable to improve existing heating furnaces in some cases.

In reply to Professor Trinks: In my paper I did not state the average coke consumption of a southern blast furnace was 2500 lb., and the table was merely introduced to show that with this particular coke consumption it was possible with steam equipment to more than supply the power requirements of steel manufacture.

In this connection attention is called to figures taken from

the statistical data published by the American Iron & Steel Institute, for all of the blast furnaces and steel plants in the United States.

Year	Production coke pig iron	Av. lb. coke per ton	Production steel ingots	Tons ingots per ton iron
1917	37,889,824	2339.7	43,619,200	1.15
1918	38,421,175	2375.2	43,051,022	1.12
1919	30,549,930	2310.2	33,694,795	1.10
1920	36,300,000	2305.7	40,881,392	1.13
Average		2342.7		1.125

From the above it will be seen that the average tons of ingots per ton of all the coke pig iron in the United States is about 1.125 instead of 1.5 as given by Professor Trinks for northern plants, and that the furnaces use about 2343 lb. coke per ton of pig iron.

A comparison of the figures given in the paper of Professor Trinks is given below per ton of pig iron, using the formula given in the paper.

	Original	Trinks	Average U.S.
Coke rate blast furnaces	2,500	1,800	2,343
B.t.u. in blast furnace gas per ton	19,635,000	12,760,000	18,080,000
B.t.u. required for furnaces	10,400,000	7,500,000	9,740,000
B.t.u. surplus blast furnace gas	9,235,000	5,260,000	8,340,000
B.t.u. coke braize (surplus)	450,000	325,000	420,000
B.t.u. surplus for power	9,685,000	5,585,000	8,760,000
B.t.u. surplus tar and by-product gas	8,685,000	6,265,000	8,150,000
B.t.u. surplus total	18,370,000	11,850,000	16,910,000
Ingots per ton pig	1.0	1.5	1.125
B.t.u. ingots required for heating total	13,000,000	19,500,000	9,040,000
" " for power	4,375,000	6,570,000	8,150,000
" " total	17,375,000	26,070,000	17,190,000
B.t.u. surplus for power	5,310,000	.....	610,000
B.t.u. deficiency for power	.....	985,000	.....
B.t.u. deficiency for heating	4,315,000	13,235,000	890,000

From the above it will be seen that for average United States conditions my statement following the table is correct. Professor Trinks' assumption of an 1800-lb. coke rate only emphasizes the necessity of improving the heating furnace efficiency if anything approaching a balance is expected.

I cannot agree with Professor Trinks that the deficiency referred to can not be wiped out, or with his inference that the use of gas engines will necessarily eliminate coal firing.

In reply to Mr. Estep: It does not make any material difference whether gas is wasted by loss at bleeders, leakage in blowing equipment or blast mains, or inefficient combustion in stoves or boilers and the purpose of the paper was, as stated, to suggest a standard for comparison of the same plant from day to day, or if possible one plant with another. Necessarily opinions differ as to the assumptions. The figure of 55 cu. ft. of blast includes an allowance of about 10 per cent for leakage which should be more ample.

I believe the formula given more closely represents the actual heat available in the gas as both the B.t.u. per cu. ft. and the cu. ft. gas per ton of pig iron increase with the coke rate.

With reference to Mr. Estep's question about power requirements, this figure will vary widely with the class of material rolled, and in the example these figures were assembled as averages of extreme values rather than from any one plant, and included all auxiliaries connected with the open-hearth and rolling-mill operations. In the example given only the surplus from the blast furnace and coke plant was considered. However, attention is called to the surplus of B.t.u. for power production of more than 100 per cent over the expected requirements, the chief deficiency in fuel for heating.

In reply to Mr. Flanagan: The installation of aspirating burners alone will not necessarily raise the efficiency of the blast furnace although it is admitted that there is probably more to be gained in the average plant by this means; as it is stated in my paper "there is generally a greater gain in efficiency possible per dollar of expenditure by efficient attention to the operation of existing equipment than by making changes in equipment without proper supervision."

From the heat-balance standpoint, dry cleaners are certainly to be preferred to wet cleaners, and some form of gas cleaner seems to be advisable for the stoves whose operation is practically continuous with the life of the furnace, but in the case of the boiler which has proper combustion chambers and facilities for cleaning the tubes; it is doubtful whether the saving justifies the necessary expenditure to clean gas for the boilers.

No. 1845

THE CONTROL OF BOILER OPERATION

SIMPLIFIED FORMULAS FOR CALCULATING CHIMNEY, COMBUSTION AND ABSORPTION LOSSES—A PROPOSED FUEL UNIT FOR BITUMINOUS COAL, ETC.

By E. A. Uehling, Milwaukee, Wis.
Member of the Society.

The problem the author sets for himself is to suggest a method that will bring complete and intelligent control of boiler operation, so far as combustion and absorption efficiency are concerned, within the horizon of comprehension and easy execution of the average operating engineer, enabling him to determine heat losses and to separate them into their several components so that he can know just where and to what extent losses occur and that he may be better able to minimize them by intelligent application of the proper remedy. The author proposes the pound-carbon fuel unit, which he shows to be practically equicalorific, and which equals 14,450 *plus* 62,000 *times the percentage of available hydrogen.*

Formulas are presented for calculating heat losses up the chimney and the combustion and absorption losses are analyzed. The application of the formulas to some scientifically conducted tests is shown.

OUR Government, through its research departments of the Bureau of Mines and the Geological Survey has made and published thousands of analyses, ultimate as well as proximate, covering all the coals of the United States, Alaska, etc. These analyses are all available and they are both interesting and instructive; but long tables of analyses are of no practical use to the operating engineer. He must have specific information of the ratio of heat input to the heat utilized or the heat wasted, it matters not which, in order to know how efficiently or wastefully his boilers are operating.

Presented at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

LIMITED USEFULNESS OF EXISTING FORMS OF FUEL DATA

2 The utter lack of direct practical usefulness of this vast storage of analytical information has been more or less keenly felt, not only by alert power-plant and combustion engineers but by the government experts themselves. A vast amount of research work has been done to determine the relation of the chemical composition of coal to its physical properties, its behavior under heat treatment, its heating value, etc. All of which is of scientific value and much of it practically useful.

3 In Bulletin 382 entitled The Effect of Oxygen on Coal, written by David White and published by the Geological Survey, we find tables of 320 analyses of air-dried coals, classified and arranged with reference to the ratio of carbon to oxygen plus ash.

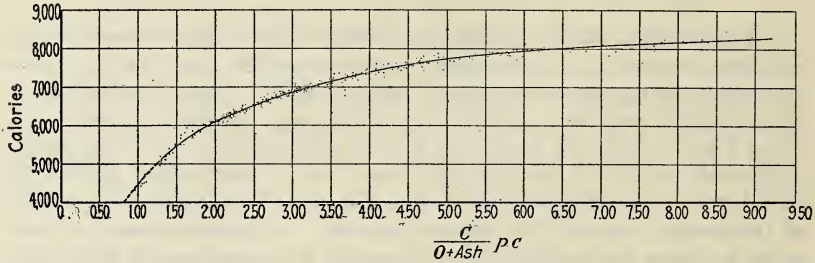

Fig. 1 Relation Between Calorimetric Value and Carbon-to-Oxygen-Plus-Ash Ratio of Coals, U. S. Geol. Survey

These analyses were made in the laboratories of the Geological Survey at Washington from samples representative of practically every variety of coal found in the United States, from anthracite down to lignite and peat. They show a variation in ash from over 28 per cent down to less than 4 per cent and in oxygen from 2 per cent to over 35 per cent. Through this investigation Mr. White has demonstrated the facts, not generally known, that oxygen and ash have practically the same negative effect on the heating value of the coal of which they are constituents so far as their heating values as determined by the calorimeter are concerned, and that it is immaterial which one of the two predominates. Fig. 1 is a reduced facsimile of the graph given in Bulletin 382 in which the relation between the calorimetric heating value and C/(O + Ash) is plotted for every sample included in the investigation. The smoothness of the curve and the small variations above and

below the average show that the relation holds practically true for all coals irrespective of the variable relation between oxygen and ash. This is the more remarkable since in many of the coals

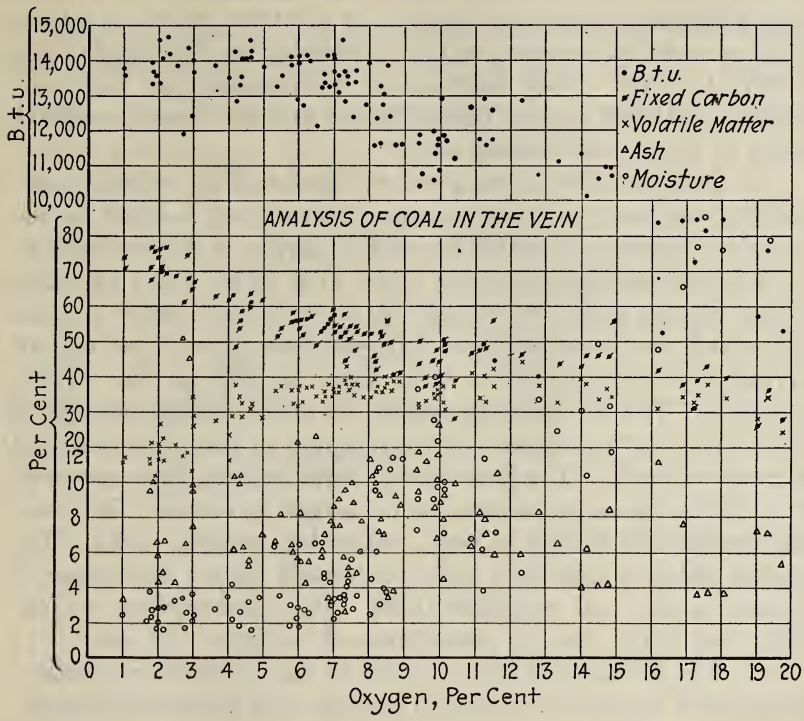

FIG. 2 RELATION OF HEATING VALUE, FIXED CARBON, VOLATILE, ASH AND MOISTURE TO PER CENT OF OXYGEN IN COMBUSTIBLE

a large part, and in lignites the larger part, of the oxygen came from the moisture retained in the air-dried samples as analyzed.

OXYGEN AN IMPORTANT CONTROLLING ELEMENT IN COAL

4 The results of Mr. White's investigation, although of undoubted scientific importance, are of no practical value to the operating engineer; but inasmuch as they greatly increase our knowledge of the properties of our coals, and may lead to something directly applicable to the problem of economic utilization of coal, and thus form an important step in that direction, they are of the greatest value. Mr. White's investigation brought out

the fact that oxygen is an important controlling element in coal, and it was therefore quite natural to attempt to establish the relation of oxygen to the other constituents of coal as well as its heating value. The analyses utilized for this purpose were taken from Bulletin 123 issued by the Bureau of Mines entitled Analyses of Mine and Car Samples of Coal, Collected in the Fiscal Years of 1913 to 1916. These analyses are representative of every important coal field in the United States and the more important mines of every coal-bearing state.

5 In Fig. 2 the heating value, fixed carbon, volatile combustible, ash and moisture contained in the coal sampled in the mines, are plotted against the per cent of oxygen in the combustible, i.e., the ash- and moisture-free coal. Due to the great variation in the ash and moisture content, the heat values occupy a broad field which runs practically level for all coals up to 8 per cent of oxygen in the combustible, then drops rapidly as the oxygen increases. The low heating values of coals having between 8 and 11 per cent of oxygen are entirely due to their high ash and moisture contents. The fixed carbon drops rapidly, then more and more gradually as the percentage of oxygen increases. The volatile combustible follows a very similar but opposite trend. The heating value of coal falls faster and faster as the percentage of oxygen in the coal substance rises. The irregularities are entirely due to the varying percentages of moisture and ash.

6 Fig. 3 illustrates the variation of the combustible constituents in their relation to oxygen in the ash- and moisture-free coals. The fixed carbon and volatile combustible follow the same trend in a somewhat narrower field than shown in Figure 2. The carbon occupies a very narrow field, and drops quite regularly nearly one-tenth of one per cent for every one per cent of increase in oxygen. The hydrogen rises rapidly but somewhat irregularly to 4 per cent of oxygen, after which it remains practically constant at 5.5 per cent up to 15 per cent of oxygen, and then drops off gradually. Sulphur, being foreign to the coal substance, varies with the ash and bears no relation to the per cent of oxygen.

7 While the interrelation of the constituents of the coal substance will no doubt prove interesting and may possibly be of some scientific worth, these graphs are of themselves of no practical value to the operating engineer, and he is the man who needs to be enlightened and helped if we are to realize the principal object

for which the Fuel Division was organized, viz., the conservation of fuel through its more economic utilization.

THE EQUICALORIFIC FUEL UNIT

8 The pound of coal as received is and will probably remain the commercial fuel unit, but because of great variation in heat-

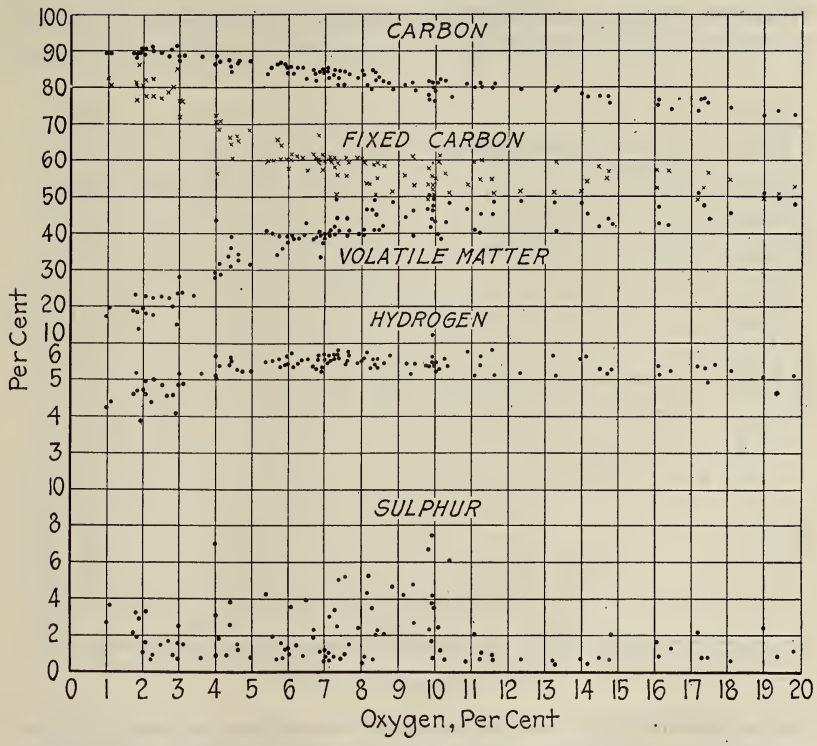

Fig. 3 Variation of Combustible Constituents in Relation to Oxygen in Ash- and Moisture-Free Coals

ing value it is a most unsatisfactory control unit. This has long been recognized by combustion experts, and the pound of combustible has been adopted as the fuel unit for comparing boiler efficiencies. This, unit, although it eliminates the variation in heating value due to the ash and moisture in coal, is practically equicalorific only between narrow limits of oxygen. As shown in Fig. 4, the heating value per pound of combustible occupies a narrow field, and drops more and more rapidly as the per cent of oxygen in

the coal substance increases. It is necessary, therefore, in order to solve our problem of making the control of boiler operation independent of the variation in the heating value of the coal as fired, to establish an equicalorific fuel unit.

9 In Fig. 4 are plotted, in their relation to oxygen, the ratio of volatile combustible to fixed carbon, (F_c/V), the hydrogen-carbon ratio, (H_a/C), i.e., the available hydrogen (H_a) per pound of carbon; the heating value per pound of combustible and the heating value of one pound of carbon and that of

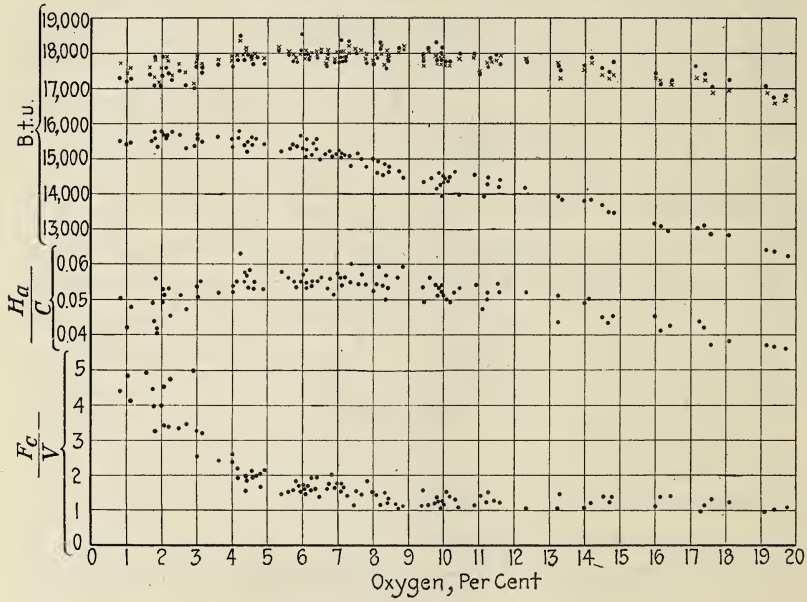

Fig. 4 Relation to Oxygen of Ratio of Fixed Carbon to Volatile, Ratio of Hydrogen to Fixed Carbon and Heating Values of Combustible and the Pound-Carbon Fuel Unit

its available hydrogen $(14{,}550 + 62{,}000H_a)$, i.e., the pound-carbon fuel unit. It will be seen that H_a is irregular up to 3 per cent of oxygen; beyond this point it covers a much narrower field, running practically level up to 12 per cent, after which it falls more and more rapidly as the oxygen increases. The trend of the field occupied by the fixed-carbon-volatile ratio (F_c/V) would indicate that 3 per cent of oxygen is the dividing line between semi-bituminous and bituminous coals. The heating

value of the pound-carbon fuel unit is stated as 14,550 + 62,000H_a B.t.u. The average weight of H_a between stated oxygen limits is 0.054 lb. per pound of carbon, hence we have 14,550 + (62,000 × 0.054) = 17,898 or say, 17,900 B.t.u. The greatest variation above or below the average value of H_a is 0.006 lb., which makes the maximum variation from 17,900 B.t.u. 0.006 × 62,000/179 = 2.08 per cent, so that for practical control purposes the pound-carbon fuel unit is essentially equicalorific. This is shown by the narrow field and practically level trend of the heat per pound of carbon

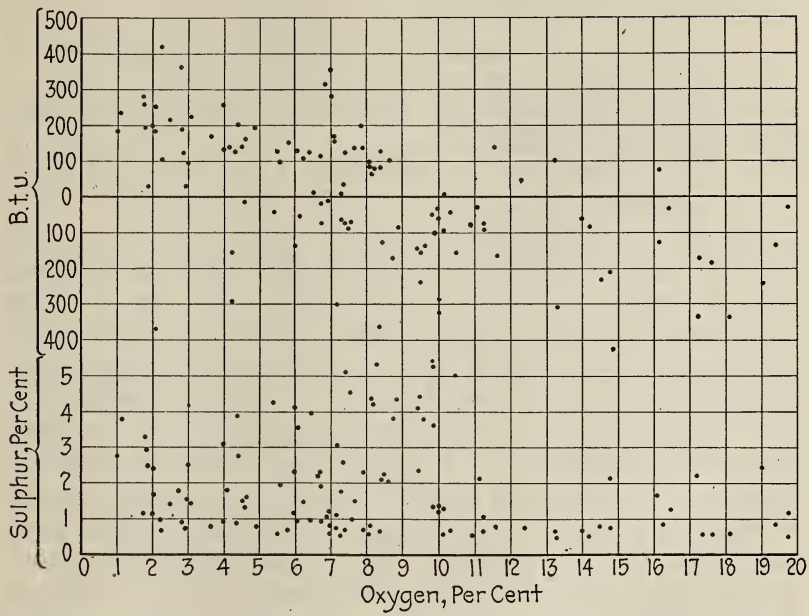

Fig. 5 Variation in Heating Values as Determined by Formula from That Determined by Bomb Calorimeter

plotted in Fig. 4. In plotting this B.t.u. field both the heating values given by the formula and those based on calorimeter determinations were used. The former are represented by crosses and the latter by dots.

10 Fig. 5 shows how the heating values calculated by the formula vary from those determined by the calorimeter. The zero variation line represents the calorimeter and the dots the variation from this when determined by the formula 14,550 + 62,000H_a. The greatest plus variation between the given oxygen limits is 350 B.t.u. and the greatest minus variation 290 B.t.u.;

the average variation either plus or minus does not exceed 1.25 per cent. The trend of the field of variation from the calorimeter is decidedly downward as the percentage of oxygen increases; below 10 per cent of oxygen the formula gives a higher and above 10 per cent a lower heating value than the calorimeter. The sulphur, which is given below, covers a broad and irregular field. The heating value of sulphur is not considered in the equicalorific fuel unit; it is therefore remarkable that the plus variations should predominate in the field opposite that of the higher percentages of sulphur. There seems to be no apparent definite relation, however.

METHODS FOR ASCERTAINING ECONOMIC OPERATION

11 There are two distinct methods for ascertaining the economic operation of a boiler:

a By determining the percentage of the heat in the fuel utilized

b By ascertaining the percentage of heat wasted.

The first is the simpler and therefore most generally practiced way. It answers well as an overall control, but does not raise the endeavor to improve boiler operation beyond the old cut-and-try methods, hence cannot lead to continuous maximum boiler efficiency. The second method, as practiced when attempted at all, is apparently more intricate and difficult than the first, and is but little understood by even the most up-to-date operating engineer. The heat losses are determined by the combustion experts for the purpose of establishing a heat balance, but heat balances mean little or nothing to the average operating engineer, and even when understood and appreciated they are of little or no help to him in controlling boiler operation. The engineer is continually confronted by contradictory advice: On the one hand, he is told that CO_2 is the all-important control factor. Keep CO_2 a maximum and heat losses will be a minimum. Carbon dioxide is the index of efficiency. On the other hand, he is admonished that CO_2 is an unreliable index; high CO_2, because of accompanying CO, is likely to cause greater heat loss than low CO_2. A third party tells him not to bother about CO_2 and CO but to keep air flow and steam flow in unison and maximum efficiency will result. Confronted by such conflicting statements, operating engineers and many power-plant managers become confused rather than enlightened.

12 Boiler efficiency rests on combustion efficiency and absorption efficiency and these are as distinct one from the other as boiler efficiency is from engine efficiency. The first is a chemical phenomenon which can be adequately diagnosed only by chemical means, and the second a physical phenomenon which must be diagnosed by physical means; without adequate diagnosis the remedy must be guessed at, and if it happens to be guessed correctly it is rarely intelligently and effectively applied. This is a scientific truism which must be borne in mind if we are to attain and maintain maximum economy in the operation of steam boilers. All other methods of control, helpful though they may be, cannot lead to maximum efficiency.

13 The heat carried to waste up the chimney can be ascertained only by analyzing the gas and measuring its temperature on leaving the boiler. To exercise intelligent control the heat in the gas must be divided into available and unavailable heat. The latter depends on the theoretical maximum percentage of CO_2 obtainable from the fuel used and the temperature of the water in the boiler. This temperature, below which the gas cannot be cooled by absorption, depends on the steam pressure which is practically constant for any given boiler, and since the maximum percentage of CO_2 is also constant, it follows that unavailable heat in the gas must be constant. The difference between the total heat and the unavailable heat is the available heat loss. The available heat loss is again composed of two parts, viz., that due to excess air and that caused by excess temperature. This analysis of the heat loss provides the means for intelligent control.

14 To enable the operating engineer to ascertain and analyze his heat losses he must know the percentage of CO_2, the temperature of the gas as it leaves the boiler and the boiler draft. These three controlling factors must be continually autographically recorded. The percentage of CO must be kept within tolerable limits, as will be shown further on. The temperature of the gas depends on:

a The rate of combustion
b The cleanliness of the heating surface
c Condition of the baffling
d The tightness of the setting.

The boiler draft is affected by all of these conditions and is therefore essential to interpret properly the cause of temperature vari-

ation. The product of the square root of the boiler draft and the CO_2 is a reliable index to the rate of combustion when the boiler and setting are in good condition.

15 Having autographic records of the essential control factors is one thing; to make effective use of the information they contain is quite another. Records of whatever kind are useless unless they are regularly scrutinized and correctly interpreted and compared and the information deduced promptly applied. To apply this information intelligently it must be put into concrete form, i.e., into B.t.u. and percentage of the heat supplied; and here is where the average operating engineer is practically helpless and it is the real cause of the adverse attitude already referred to, his aversion to scientific methods of control. To overcome this attitude heat-loss calculations must not only be brought within the horizon of his ready comprehension, but they must be made so easy that he can make them with the least trouble and expenditure of time.

THE AUTHOR'S HEAT-LOSS FORMULAS

16 In an appendix to his paper on the Physical and Chemical Control of Boiler Operation,[1] the author deduced a set of formulas for calculating the heat losses up the chimney, which are again presented below in slightly modified form. These formulas attracted no attention at the time nor have they since, at least no comment upon them has come to the notice of the writer. The reason for this may have been that they are based on the pound-carbon fuel unit, the advantages of which were not then apparent.

17 The formulas are as follows:

$L_d = (0.24 + 58.46/CO_2) \times (T - t)$

$= $ B.t.u. carried to waste by the dry gas ..... [1]

$L_c = 10{,}150\ CO/(CO + CO_2)$

$= $ B.t.u. loss due to CO contained in gas ..... [2]

$L_h = H_t \times [(10{,}642 - 9t) + 4.3\ (T - 212)]$

$= $ B.t.u. carried to waste by the steam contained in the gas;

[1] Trans. Am. Soc. M. E., vol 40, p. 714. A number of typographical errors remain uncorrected but these will be readily detected by anyone taking the trouble to follow out the deductions.

H_t = weight of total hydrogen per pound of carbon in the coal, including the moisture ..... [3]

$$A_e = \frac{2100}{(1+3H_a)\times CO_2} - \frac{100+238H_a}{1+3H_a}$$

= per cent excess air ..... [4]

$O_c = 21 - CO_2 \times (1+2.38H_a)$

= per cent excess oxygen ..... [5]

$MCO_2 = 21/(1+2.38H_a)$

= maximum per cent of CO_2 obtainable from the fuel burned ..... [6]

$A_w = 243.6/CO_2 + 8H_a$

= pounds of air supplied per unit of fuel consumed .... [7]

$G_w = 1 + 243.6/CO_2$

= pounds of dry gas produced ..... [8]

$A_v = 3192/CO_2$

= cu. ft. of dry gas at 62 deg. fahr. ..... [9]

$S_w = 9H_t$

= pounds of steam in gas. ..... [10]

Z = square root of the boiler draft. ..... [11]

$Z \times CO_2$ = index of rate of combustion. ..... [12]

All the formulas are based on the pound-carbon fuel unit.

THE CONTROL FACTORS

18 To establish a heat balance, the B.t.u. supplied by the air must be deducted from the heat in the gas, hence the temperature factor $(T - t)$ must be used in Formulas [1] and [3], but as already stated, heat balances are of uo use for direct, prompt, and continuous control. To enable the operating engineer to exercise effective control over combustion and absorption efficiency, he must know not only how much heat is carried to waste by the gases, but how much of this heat is available to the boiler and what part of the available heat loss is chargeable to the process of combustion and how much to absorption; and since all the heat brought in by the air is contained in the unavailable part of the heat in the gas, the temperature of the air may be ignored, thus simplifying the formulas by eliminating a variable term. The calculations for determining the two essential control factors are:

$A = (0.24 + 58.46/CO_2) \times T =$ total B.t.u. in the dry gas, and

$B = (0.24 + 58.46/MCO_2) \times T_w =$ unavailable B.t.u. in the dry gas, in which $T_w =$ temperature of water in boiler;

then

$A - B =$ the theoretically available heat carried to waste by the dry grass. In formula [3] the term H_t $(10{,}642 - 9t)$ represents latent heat which is all unavailable, hence if we let —

$C = (4.3H_t) \times (T - 212)$ — total sensible heat of H_2O in the gas, and

$D = (4.3H_t) \times (T - T_w) =$ unavailable sensible heat in H_2O in gas, then

$C - D =$ available heat in H_2O contained in the gas, hence,

$(A+C) - (B+D) = E =$ total available heat carried to waste up the chimney

The heat loss E is partly due to combustion and partly to absorption inefficiency.

COMBUSTION AND ABSORPTION LOSSES

19 Combustion would be 100 per cent efficient if all the combustible elements in the fuel were completely oxidized in the furnace, using the theoretical weight of air required, and absorption would be 100 per cent efficient if it reduced the temperature of the gas resulting from 100 per cent efficient combustion to the temperature of the water in the boiler. The component parts of combustion losses are:

a The heat loss due to excess air.

b The heat value of the unburned combustible in the gas $(CO + C_x + H_y)$ and

c The carbon in the ash.

The component parts of the absorption losses are:

a Heat loss due to dirty heating surface.

b Heat loss due to defective baffling, and

c Heat loss due to excessive driving.

All these component parts of the waste of available heat can be determined from the interrelation between the three essential autographic records, for any particular time or any desired period, such as a test, a day or a shift, as follows:

20 Applying our formula we have:

$a = [(0.24 + 58.46/CO_2) - (0.24 + 58.46/MCO_2)] \quad T =$ B.t.u. loss due to excess air

The combustible constituents in the gas are CO and an undetermined weight of $(C_x + H_y)$. The heat value of the latter probably never exceeds 0.2 of the former in boiler flue gas, and is negligible when CO is kept within proper limits.

$b = 10{,}150\ CO/(CO+CO_2) =$ B.t.u. loss from CO.

c The B.t.u. loss due to unburned carbon must be determined by analysing the ash. We have then

$C_{ef} = 100 - (a+b+c)/179 =$ combustion efficiency.

The component parts of heat lost to absorption can be located on the autographic charts; but can be quantitatively determined by our formula only in total; thus:

$$A_{ef} = 100 - \frac{(0.24+58.46/MCO_2) \times (T - T_w)}{179} = \text{absorption efficiency.}$$

The component parts of absorption loss may, however, be approximately determined, as will be shown by example further on.

SIMPLIFICATION OF FORMULAS

21 All this may look like a rather formidable array of equations to place before the average operating engineer, with the expectation that he apply them successfully. Such expectation would be futile. The author, however, is not now confronting average operating engineers; he imagines he is addressing scientific as well as practical fuel conservationists and combustion experts, who will understand that the foregoing formulas are applicable to all commercial fuels from coke to natural gas, whereas operating engineers rarely have to deal with more than one kind of fuel at a time. Limiting ourselves for the present to bituminous coal, these formulas, being based on the equicalorific fuel unit, can be greatly simplified and those essential for control reduced to few in number.

22 In Formula [1], the factor $(0.24+58.46/CO_2)$ is equal to the B.t.u. per degree of temperature contained in the weight of dry gas produced in burning one unit of fuel. Letting X equal this value, then $XT =$ total B.t.u. in dry gas per unit of fuel. Substituting the average value of H_a $(= 0.054)$ in Formula [6], we have $MCO_2 = 21/1.128 = 18.6$ per cent $=$ the theoretical maximum percentage of CO_2 obtainable from bituminous coal. Using this value of CO_2 in Formula [I] we have $(0.24+58.46/18.6) \times T_w = 3.38T_w =$ B.t.u. of unavailable heat in dry gas. T_w

varies with steam pressure, but since this is practically constant for any given boiler plant, $3.88T_w$ is also constant. $XT - 3.38T_w$ = B.t.u. = available heat. Now since X represents the total B.t.u. per degree of temperature in the weight of dry gas resulting from the combustion of one unit of fuel burned with an excess of air, and 3.38 = B.t.u. = the heat per degree of temperature in the weight of gas that results when a unit of fuel is burned with the theoretical weight of air required for the complete combustion, we have $(X - 3.38)T$ = B.t.u. of available heat wasted because of excess air and $3.38\ (T - T_w)$ = B.t.u. of available heat wasted due to excess temperature.

23 In Formula [2], $10{,}150\ CO/(CO+CO_2)$, the factor $CO/(CO + CO_2)$ represents the weight of carbon burned to CO. This formula can be somewhat simplified by letting $10{,}150 \times CO = Y$ so that $Y/(CO+CO_2)$ = B.t.u. wasted due to CO.
Our essential control formulas are then:

XT = total B.t.u. contained in dry gas. ..... [A]
$3.38\ T_w$ = B.t.u. unavailable in dry gas. ..... [B]
$XT - 3.38T_w$ = B.t.u. available in dry gas. ..... [C]
$(X - 3.38)\ T$ = B.t.u. loss due to excess air. ..... [D]
$3.38\ (T - T_w)$ = B.t.u. loss due to excess temperature. .. [E]
$Y/(CO+CO_2)$ = B.t.u. loss due to CO in gas. ..... [F]
$Z \times CO_2$ = index to rate of combustion. ..... [G]

The value of $3.38T_w$ is practically constant for any given power plant. The values of X, Y and Z may be read off directly from tables prepared for the purpose or they may be found from curves as shown in Fig 6. Only a fraction of the B.t.u. in the steam contained in the gas is sensible heat and the greater part of this is unavailable to the boiler. The heat loss represented by Formula [3] therefore has no control value and may be ignored. By substituting the value of $H_a = 0.054$ in Formulas [4], [5], and [6], they reduce respectively:

$A_c = 1810.35/CO_2 - 97$ = per cent excess air ..... [4a]
$O_c = 21 - 1.13CO_2$ = per cent excess oxygen. ..... [5a]
$MCO_2 = 18.6_2$ = maximum per cent CO_2..... [6a]

The information derived from [4a] and [5a], though not essential for control purposes, may be desirable to have.

24 Combustion and absorption efficiency are the two factors on which economical boiler operation depends. Maximum absorp-

tion efficiency in any given boiler is maintained by keeping the boilers clean inside and out, and maintaining the baffling and setting in perfect condition. The former is a matter of routine conscientiously performed and the latter can be secured by proper and frequent inspection. If the proper tools are supplied, maintaining absorption efficiency is a simple matter. Combustion efficiency, on the other hand, is an entirely different proposition and cannot be maintained by any definite predetermined routine of operation. It depends on numerous and frequent adjustments: The rate of

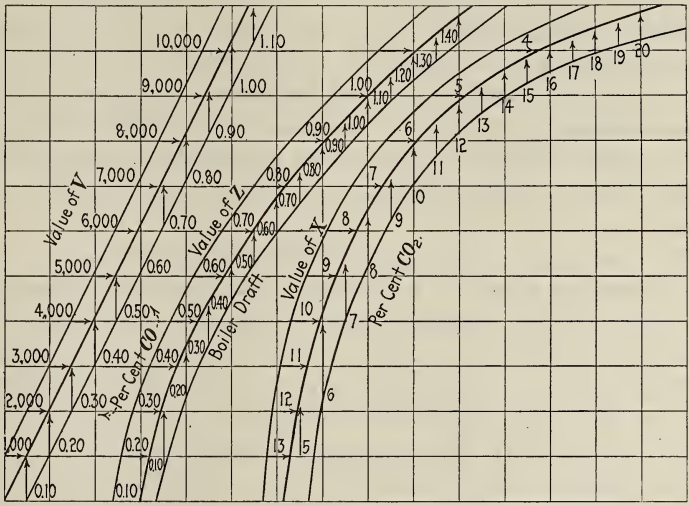

Fig. 6 Values of X, Y and Z

combustion must be adjusted to the steam demand. The draft must be adjusted to maintain the rate of combustion, the coal supply must be kept in harmony with the draft, and the firebed must be kept in condition to make the draft economically effective. The air supply must be adjusted so that the proper proportions enter, respectively, below and above the firebed. None of these adjustments can remain fixed but must be frequently changed as a whole and in relation to one another if maximum combustion efficiency is to be achieved.

25 It is not possible to maintain maximum combustion efficiency by inspection. Without the aid of the proper and necessary means for getting the essential data, boiler operation is necessarily

a more or less haphazard performance, depending on intuition and guessing. To achieve the best results in any undertaking proper instruments must be provided. This is as true of operating boilers as it is of navigating a ship or conducting a scientific research. Suppose that in addition to the steam gage and water glass, which are always supplied, (because required by law) the necessary instruments, viz., an Orsat, a continuous recording CO_2 meter and indicator, a recording pyrometer and a boiler draft gage have been properly installed. The first and most important step necessary to insure success is to put some one in charge and hold him strictly responsible for the continuous and correct functioning of the instruments. The principal cause for failure of such instruments is due to overlooking or neglecting this necessity. A further reason why so many installations of these instruments have not led to the results predicted or anticipated is that the operating engineer has had no simple means for getting the information autographically recorded into the form necessary for direct and intelligent application.

26 The object of the simplified control formulas is to bring the mathematics necessary to determine and locate the losses within easy reach of the average operating engineer, and especially to enable him to make the necessary calculations with the least expenditure of time and trouble. The fact that these formulas are based on the equicalorific fuel unit gives them a much greater practical value than their simplicity would indicate. It makes effective control of boiler operation quite independent of the analysis and heating value of the coal as fired. If the necessary equipment is provided, any up-to-date operating engineer should by the aid of such equipment and the simplified formulas be able to maintain the highest combustion and absorption efficiency that the structural and operating conditions of his plant (including the quality of the fuel used) will permit, and if maximum combustion and absorption efficiencies are maintained, minimum waste of fuel will necessarily result.

27 Perhaps enough has been said to make clear that in the operation of boilers, the best results can be attained and maintained only when the operating engineer can know and does know what results he is getting at any time and all the time, also that he should have an ideal to strive for, a bull's eye to aim at. None the less it may not be amiss to show how these formulas prove

out by applying them to a few authoritative tests, in order to strengthen the faith of the many "doubting Thomases."

APPLICATION OF FORMULAS

28 Tables 1 and 2 show the application of the heat-loss control formula to ten scientifically conducted tests. These tables

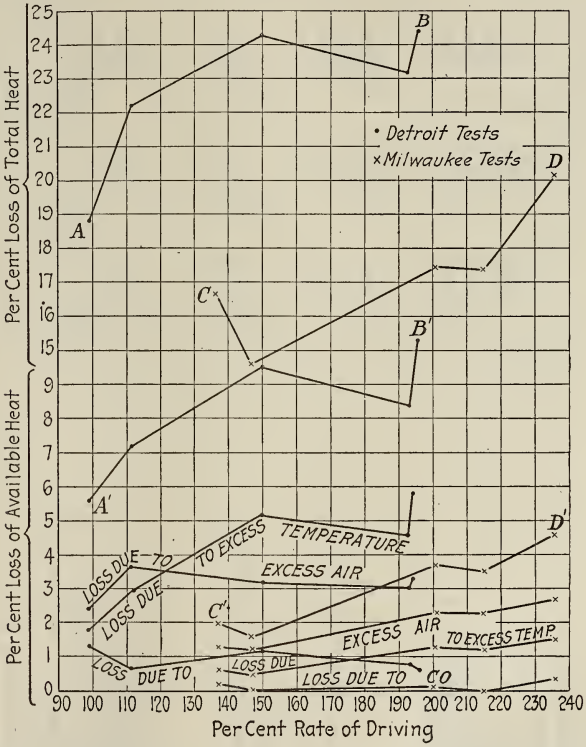

FIG. 7 GRAPHICAL REPRESENTATION OF APPLICATION OF HEAT-LOSS CONTROL FORMULAS

are self-explanatory. The tests treated in Table 2 were taken from the series of tests made of the large boilers of the Delray Plant of the Detroit Edison Company by Dr. Jacobus[1] as reported to this Society in the paper which he presented at the Annual Meeting in December, 1911. Table 1 treats of the pulverized-coal tests made by Henry Kreisinger of the boilers of the Lakeside Power

[1] See TRANS., vol. 33, p. 565.

TABLE 1 UEHLING HEAT-LOSS CONTROL FORMULAS APPLIED TO THE PULVERIZED-COAL BOILER TESTS OF THE LAKESIDE POWER PLANT, MILWAUKEE

	Data Essential for Control				
Number of test	1	4	3	2	5
CO_2 = Per cent carbon dioxide in gas	15.8	16.0	14.7	14.6	14.1
CO = Per cent carbon monoxide in gas	0.05	0.00	0.03	0.00	0.09
T = Temperature of gas on leaving boiler, deg. fahr.	434	430	482	475	490
T_w = Temperature of water in boiler, deg. fahr.	409	409	411	411	411
B_d = Boiler draft, i.e., resistance to gas through boiler, in water	0.30	0.355	1.03	0.92	1.2
RESULTS OF TESTS					
Pounds of coal consumed per hour	5980	6650	9560	9740	11350
Units of fuel consumed per hour	4114	4319	6331	6545	7424
Per cent rate of driving	137	147	209	215	236
Per cent boiler efficiency	83.3	85.4	82.5	82.6	79.8
CONTROL FACTORS OBTAINED FROM FIG. 5					
X = B.t.u. in dry gas per unit of fuel and degree of temperature	3.94	3.9	4.17	4.24	4.38
Y = 10,150 CO	508	0.0	344	0.0	914
Z = Square root of boiler draft (B_d)	0.55	0.59	1.01	0.957	1.09
ANALYSIS OF HEAT LOSSES					
XT = B.t.u. of total loss of sensible heat in dry gas	1710	1673	2034	2014	2177
XT_w = B.t.u. of unavailable sensible heat in dry gas	1382	1382	1389	1389	1389
$XT - 3.38T_w$ = B.t.u. of available sensible heat in dry gas	328	291	645	625	788
$(X - 3.38)T$ = B.t.u. loss of available heat due to excess air (a)	230 = 1.3%	219 = 1.2%	405 = 2.3%	408 = 2.3%	490 = 2.7%
$3.38\,(T - T_w)$ = B.t.u. loss of available heat due to excess temperature (b)	96 = 0.53%	68 = 0.41%	240 = 1.3%	217 = 1.2%	277 = 1.55%
$Y/CO + CO_2$ = B.t.u. loss due to CO in gas (c)	32 = 0.18%	00 = 0.0%	20 = 0.11%	0.0 = 0.0%	64 = 0.36%
$a + b + c$ = B.t.u. total loss of available heat in dry gas	358 = 2.0%	287 = 1.6%	665 = 3.7%	625 = 3.5%	831 = 4.6%
$100 - (a + c)\%$ = Combustion efficiency, per cent	98.5	98.8	97.6	97.7	96.9
$100 - b\%$ = Absorption efficiency, per cent	99.47	99.6	98.7	98.8	98.45
$100 - (a + b + c)\%$ = Combined absorption and combustion efficiency, per cent	98.0	98.4	96.3	96.5	95.4
ZCO_2 = Index of rate of combustion	9.16	9.53	14.7	14.6	15.44

TABLE 2 UEHLING HEAT-LOSS CONTROL FORMULAS APPLIED TO TESTS OF LARGE BOILERS OF THE DETROIT EDISON COMPANY

	Data Essential for Control				
Number of tests to which formulas were applied	5	3	6	17	18
CO_2 = Per cent carbon dioxide in gas	14.4	13.5	14.66	14.69	14.16
CO = Per cent carbon monoxide in gas	0.35	0.18	0.31	0.20	0.16
T = Temperature of gas on leaving boiler, deg. fahr.	483	542	662	636	694
T_w = Temperature of water in boiler, deg. fahr.	387	387	388	390	390
B_d = Boiler draft, i.e., resistance to gas through boiler, in water	0.08	0.18	0.31	0.61	0.78
RESULTS OF TESTS					
Pounds of dry coal consumed per hour	6606	8130	11295	13761	14987
Units of fuel consumed per hour	5083	6402	8963	11163	12480
Per cent rate of driving	94.0	113.8	150.7	193.3	195.7
Per cent boiler efficiency	81.15	77.45	75.28	76.73	75.57
CONTROL FACTORS OBTAINED FROM FIG. 5					
X = B.t.u. in dry gas per unit of fuel and degree of temperature	4.3	4.6	4.23	4.22	4.37
Y = 10,150 CO	3510	1850	3225	2000	1600
Z = Square root of boiler draft (B_d)	0.28	0.42	0.555	0.78	0.88
ANALYSIS OF HEAT LOSSES					
XT = B.t.u. of total loss of sensible heat in dry gas	2077	2493	2800	2684	3033
$3.38T_w$ = B.t.u. unavailable sensible heat in dry gas	1308	1308	1311	1318	1318
$XT - 3.38T_w$ = B.t.u. loss of available sensible heat	769	1185	1483	1366	1725
$(X - 3.38)T$ = B.t.u. loss due to excess air (a)	444 = 2.4%	661 = 3.7%	563 = 3.2%	534 = 3.0%	687 = 3.8%
$3.38\,(T - T_w)$ = B.t.u. loss due to excess temperature (b)	325 = 1.8%	524 = 2.9%	926 = 5.2%	832 = 4.6%	1038 = 5.8%
$Y/CO + CO_2$ = B.t.u. loss due to CO in gas (c)	232 = 1.3%	111 = 0.62%	215 = 1.2%	134 = 0.75%	119 = 0.66%
$a + b + c$ = B.t.u. total loss of available heat in dry gas	1001 = 5.6%	1296 = 7.2%	1704 = 9.5%	1500 = 8.4%	1844 = 10.3%
$100 - (a + c)\%$ = Combustion efficiency, per cent	96.3	95.7	95.6	96.25	95.7
$100 - b\%$ = Absorption efficiency, per cent	98.2	97.1	94.8	95.4	94.2
$100 - (a + b + c)\%$ = Combined combustion and absorption efficiency, per cent	94.4	92.8	90.5	91.6	89.7
ZCO_2 = Index of the rate of combustion	3.74	5.67	8.14	11.46	12.46

Plant of the Milwaukee Electric Railway and Light Co. as reported at the Spring Meeting, 1921. All the five tests reported are represented.

29 The results of applying the heat-loss control factors are graphically shown in Fig 7. The graph *AB* represents the percentage of total heat wasted, i.e., 100 minus the per cent efficiency as found by tests of the Detroit Edison Company's boilers; and *CD* represents the total heat wasted as found by test of the

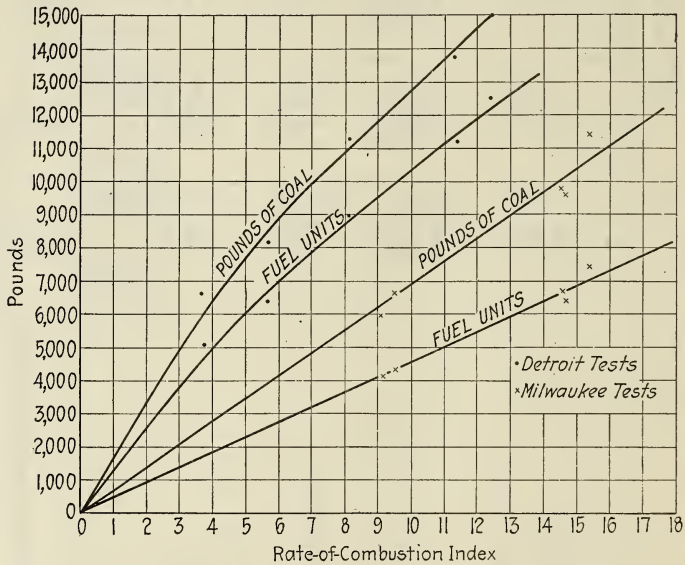

Fig. 8 Relation of Rate of Combustion to Combustion Index, ZCO_2

Milwaukee Electric Railway and Light Co.'s boilers. *A'B'* and *C'D'* show the respective losses of available heat of these two sets of tests as determined by the control formulas. The other graphs show where and to what extent the loss of available heat took place. Comparing graphs *AB* and *CD* respectively with *A'B'* and *C'D'*, it will be seen that they are quite similar in contour, and this close similarity demonstrates the reliability of the loss of available heat as an index of boiler efficiency. The reliability of this index can be disturbed only by excessive moisture in the coal or excessive loss of carbon through the grate. The former adds to the unavailable heat and hence cannot be con-

trolled by the operating engineer, and much less by the fireman; the latter can be detected and roughly estimated by observation and promptly remedied. Fig. 8 shows the rate of combustion in its relation to the combustion index ZCO_2. As the graphs show, this necessarily varies with the type and size of boiler, the arrangement of baffling, etc. The characteristic of the graph must be determined for each type and construction of boiler. This done, it becomes a practically reliable check on the effort

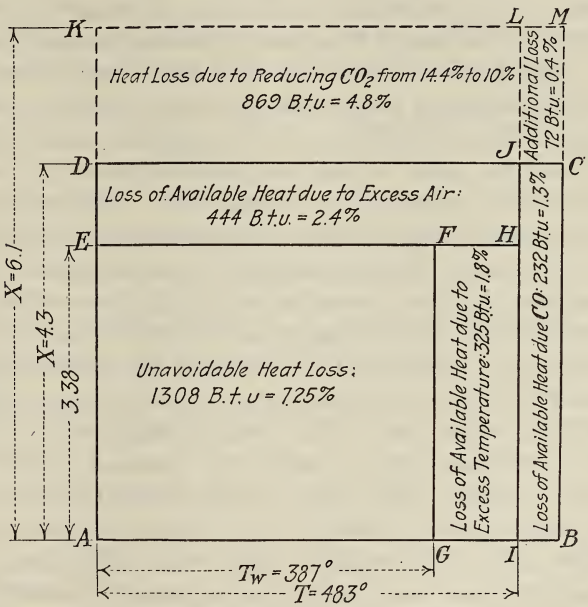

Fig. 9 Heat-Loss Diagram

of the fireman to burn his proper share of coal in keeping up the steam pressure. Since the percentage of CO_2 is the principal factor in determining combustion efficiency as well as this index, the inefficient fireman is compelled to work harder to make his proper share of the steam.

30 The loss of available heat up the chimney, as is shown by the tables and graphs, is due to three causes: excess air, excess temperature and CO. The components of this heat loss are shown by the tables and in Fig. 7. These losses can be quantitatively illustrated as shown in Fig. 9. The rectangle *ABCD* represents the

total heat carried to waste by the dry gas, and *AGFE* the heat unavailable to the boiler. *EHJD* represents the loss of theoretically available heat due to excess air, *GIHF* that due to excess temperature and *IBCJ* to CO. Now supposing that the excess air had been increased so as to reduce the CO_2 to 10 per cent, the loss of available heat would have been increased by *DCLK;* assuming the temperature to remain the same and the percentage of CO also unchanged the loss due to the latter would be increased as shown by *CJML*. But supposing that the CO had been eliminated by this reduction in CO_2, the gain would have been 1.3 per cent and the loss 4.8 per cent, resulting in a net loss of 3.5 per cent. This is not the place to discuss this phase of the problem, except to call attention to the fact that care must be exercised in eliminating CO at the expense of CO_2, since the loss is liable by far to exceed the gain. Complete combustion is by no means synonymous with most efficient combustion. The greatest practical value of this method of control consists in its simplicity as well as its adequacy. Three autographic records preferably on the same chart are all that is required to furnish the necessary data. The calculations are of the simplest nature and the method is quite independent of the analysis of the coal and its variable heat value as long as it is true bituminous coal, i.e., with oxygen within the limits of 3 to 12 per cent. The heat brought in by the air is contained in the unavailable heat, hence does not affect the reliability of the control. The loss of available heat is the only loss that need be considered because it is the only loss that is controllable. This method of control fixes the responsibility, inasmuch as it separates the heat loss into two parts, namely, combustion loss controllable by the fireman, and absorption loss controllable by the boiler cleaners.

31 This may be a bold statement considering the present general state of the art and science of boiler operation, but the author hopes and believes that this method of control is a step in the right direction. If it is not, he wants to be set straight; if the method is fallacious this is the time and place to disprove the claim he makes for it. So far as the equicalorific fuel unit is concerned, the author has no fear that it will be upset. There may be exceptional coals within the prescribed oxygen limits that will not come within the practically allowable limits of the equicalorific heating value, but there will be few if any. The average analysis

of the coal used in the five Detroit tests varied 4.75 per cent in ash, 2 per cent in oxygen and 804 B.t.u. in heating value while the separate analyses of the same coal varied as much as 1000 B.t.u., whereas the variation in the heating value of the pound-carbon fuel unit as determined from the average analysis of the coal used in these tests differs from the equicalorific heating value by only 372 B.t.u. The heating value of the coal used in the five Milwaukee tests varied 834 B.t.u., the oxygen not being given in the analysis. The variation in the pound-carbon fuel unit could not be determined but as found from the analyses of representative samples of Illinois coal, the pound-carbon fuel unit is practically equicalorific. The highest heating value of the coal used in the Detroit tests was 14,493 B.t.u. and the lowest of that used in the Milwaukee tests was 11,483 B.t.u., a difference of over 3000 B.t.u., but, as we have seen, this difference in heating value per pound of coal does not affect the constancy of the equicalorific fuel unit nor the practicability of the heat-loss control formulas based thereon.

32 The author has placed emphasis on the necessity of getting the attitude of mind of the operating engineer in line with the endeavor of the Fuels Division to reduce the consumption of fuel in the production of steam to a minimum. To make worthwhile progress there must be coöperation from the fireman up to the owner or managing director of the steam-power-producing and coal-consuming plants. The urge for the necessary equipment should come from below and be granted from above. To this end education is necessary at the top and bottom as well as in between. But who is to do the educating? Engineering magazines do much. Their articles on the subject are good, bad and indifferent, often contradictory and therefore more or less confusing. The instrument makers send out a lot of valuable information, but it is too limited in scope and exaggerated in tone. Their agents do considerable educational work but it is looked upon as being mercenary and is therefore not effective. Consulting engineers and combustion experts invade power plants, and generally with good results; not infrequently, however, they merely make elaborate boiler tests, establish a scientific heat balance, give numerous practical and impractical instructions, take their fee and go on their way rejoicing. The operating engineer, who is frequently not taken into their confidence, knows but little more after they have left than he did before they came, and consequently permanent operating improvement does not result.

33 The ten boiler tests illustrating the application of the author's control formulas show what can be accomplished in the way of economic boiler operation. The average efficiency of boilers in plants the country over is from 15 to 20 per cent below these high standards. The reasons for the slow progress, in the author's judgment, have been: (1) Lack of fundamental control principles which can be readily understood and easily applied by the operating engineer; (2) lack of knowledge and appreciation, by the owners and managers, of the preventable waste of fuel in their plants; (3) lack of coöperation between instrument maker and the purchaser of such instruments; (4) lack of equipment of the necessary instruments; and (5) last but not least, lack of proper instruction to enable the operating engineer to coördinate and interpret the information autographically recorded when such instruments are supplied.

DISCUSSION

A. T. Hutchins. In connection with the formulas which Mr. Uehling shows for checking boiler performance, referring especially to flue-gas analyses, it may be interesting to note a very simple graphic means for checking precision of work done with an Orsat in sampling flue gas from boilers. The accompanying chart, Fig. 10, shows three lines: The broken line is drawn from formula [6a]. The solid line is drawn to average points plotted from flue-gas analyses published from Detroit Edison Co.'s tests of 1910 and 1911. This line is extended through the base line at 20.7, this scale being per cent of oxygen shown in analyses of flue gas. The dash-dot line is drawn to average points plotted from flue-gas analyses published from New York Edison Co.'s tests. This line is extended in the same way.

In studying this chart it is evident that with 20.7 per cent of oxygen in the atmospheric air, when the gases resulting from combustion are reduced to zero, analyses will show the total per cent of oxygen found in atmospheric air. As the different stages of combustion efficiency are reached points showing per cent of oxygen and CO_2 in the flue gas plotted on this chart will lie along a straight line if combustion is complete. That is, if there is no unburned hydrogen, hydrogen-carbons, or carbon monoxide present, and the proportion of fixed carbon and volatile matter consumed

remains the same. This will hold true for any one particular fuel whether bituminous coal, anthracite, or natural gas.

Of course, this chart can not be used with blast-furnace gas having varying amount of CO_2 present in the gas before burning.

The precision of the gas analyses made in Detroit Edison Co.'s tests is very interesting; the variation in the other points plotted doubtlessly could be explained with data which are not at hand.

We suggest that such a chart can be used by the average man working with combustion problems with resulting increased interest and accuracy in his work.

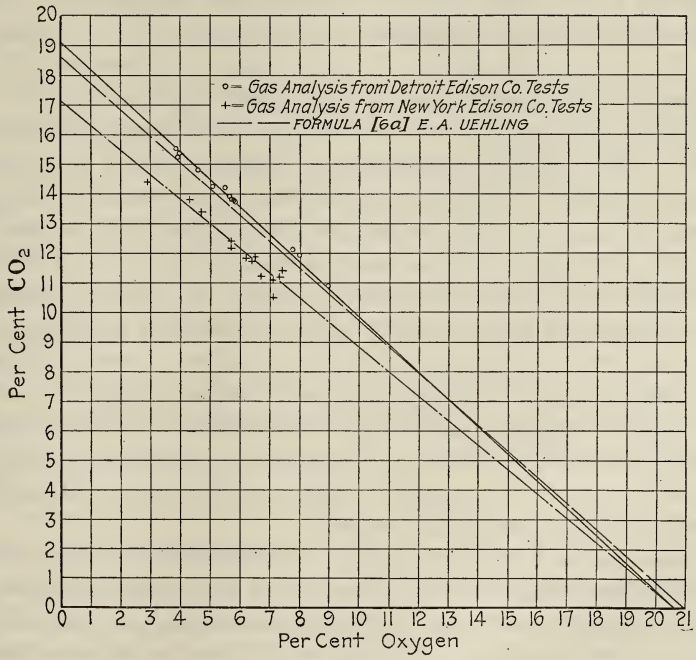

Fig. 10 Comparison of Test Values of Gas Analyses with Author's Formula

Nevin E. Funk. In Par. 26 the author says: "If the necessary equipment is provided, any up-to-date operating engineer should by the aid of such equipment and the simplified formulas be able to maintain the highest combustion and absorption efficiency that the structural and operating conditions of his plant (including the quality of the fuel used) will permit, and if maximum combustion and absorption efficiencies are maintained, minimum waste of fuel will necessarily result."

That can not be done hour by hour or minute by minute by the use of the scheme he has outlined. Of course, it is of some value to know the next day what you have done, but it is far better to arrange the control equipment so that it is possible to know that the coal is burned efficiently at the moment of its burning, and the writer does not believe this scheme has done that.

In Par. 24 the author says: "If the proper tools are supplied, maintaining absorption efficiency is a simple matter." That is a very good statement. The writer cannot see, however, that Mr. Uehling has stated the proper tools in this paper. There is an attempt at a statement in Par. 25 where the author says: "in addition to the steam gage and water glass, which are always supplied (because required by law), the necessary instruments, viz., an Orsat, a continuous recording CO_2 meter and indicator, a recording pyrometer and a boiler draft gage have been properly installed." These are the instruments he needs to check this boiler efficiency scheme. It will be noticed there is no output measurement on the boiler. If the operator does not know the output of the boiler, how does he know the correct draft? If he does not know the correct draft, the results will be hit or miss. He guesses at it by looking at the fire, and if there is more than one boiler, or if there are several men running a bank of boilers, there is a probability that one man may keep a splendid fire and make the other fellow take the ragged ends, and so it seems that a very important instrument has been left out of the author's statement, that is, a steam-flow meter or a water-flow meter, which shows the output the boiler is delivering every instant.

Then, to match up the readings of some of these meters, such as a draft gage, there must be a table on the boiler telling what the draft should be for each output, or some other scheme should be used, to give a man an indication of the relation between steam flow and draft.

These comments have been in a way destructive, and the writer does not want to leave the impression that he feels that this paper is not important because he thinks it is, but it is not a means for the boiler operator to control the boiler.

F. G. Cutler. The author has apparently overlooked one phase that applies to steel-plant operations, particularly in operation with blast furnaces or by-product gases. With by-product

gas the best CO_2 content is around 7 per cent, whereas in the case of coal it is 16 per cent; and with blast furnace gas it is from 22 to 25 per cent, so that with two fuels burning on the same boiler at different times or at the same time, the CO_2 does not give a correct measure of efficiency.

THE AUTHOR. Mr. Funk's criticism, that a steam-flow meter or water meter is not included among the instruments designated as necessary for the control of combustion and absorption efficiency overlooks the fact that the product of the square root of the boiler draft and the per cent of CO_2 ($\sqrt{Bd}$ x CO_2) is a sufficiently accurate index of the rate of combustion to detect a loafing or inefficient fireman.

While the steam-flow meter or water meter will show the lazy fireman they give no clue whatever as to his efficiency. The steam-flow meter is a very desirable instrument to have but it is of no immediate value as an efficiency control. Neither is it a true indication of the draft necessary to burn the coal at the rate required to make the boiler produce its full quota of the steam demand, because the rate of combustion for a given draft necessarily varies with the thickness of the fire-bed as well as its condition, etc.

A steam-flow meter on a boiler shows its rate of steam production, which is a very desirable thing to know, but it is not an essential factor in the control of combustion and absorption efficiency.

In the per cent of CO_2 we have the prime index of combustion efficiency, in the temperature of the escaping gas we have the index of absorption efficiency and in $\sqrt{Bd}$ x CO_2 an index of the rate of combustion. The author had hoped to have demonstrated that these three factors are *essential* and *adequate* for the effective and intelligent control of boiler efficiency. It goes without saying that the gas must be tested for CO in order to establish the maximum economic per cent of CO_2, which will vary with the type of furnace and the kind of fuel used.

In reply to Mr. Cutler's criticism the author would say that the general formulas apply to all natural fuels and perhaps also to artificial fuels containing no CO_2, or CO or free oxygen. Blast furnace and producer gas have not yet been investigated by the author, but it does not appear likely that the formulas as developed will apply to them.

No. 1846

BOILER-ROOM PERFORMANCE AND PRACTICE OF COLFAX STATION, DUQUESNE LIGHT COMPANY

By C. W. E. Clarke, New York, N. Y.
Member of the Society

The paper describes in detail the operation of the boiler plant of the Colfax Station of the Duquesne Light Co., located at Cheswick, Pa. There are included sample analyses of fuel and ash, the piping schedule, temperature and CO_2 records, typical water analyses, observations of dissolved oxygen in boiler feedwater, boiler-test data and a typical heat balance covering four months.

IT IS the purpose of this paper to describe in some detail the operation of the boiler plant in the Colfax Station of the Duquesne Light Co., located at Cheswick, Pa. Fig. 1 is a cross-section of the boiler house and part of the turbine room, and Fig. 2 is a general view of the exterior of this plant. The Colfax Station at present contains one three-element compound Westinghouse turbine of 60,000 kw. capacity and an additional similar machine will be installed this summer. The present plans for this station contemplate an ultimate capacity of 360,000 kw. The boiler plant contains seven boilers, 18 tubes high by 51 wide, each containing 20,867 sq. ft. of heating surface; and four additional boilers, 20 tubes high by 51 wide, each containing 22,914 sq. ft. of heating surface, are being installed this summer. The station is designed for base-load operation. Under present conditions it is possible to maintain a load in excess of 50,000 kw. for most of the day. For the night periods, from about midnight until seven or eight in the morning, the load may fall as low as 30,000 kw.

2 The boiler room was designed with a view to securing the highest ultimate efficiency in point of fuel, labor and fixed charges. It will be noted that the labor per horsepower is very small. The

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area covered by the boiler house per boiler horsepower is only 1.37 sq. ft.

FUEL AND FUEL HANDLING

3 Coal is brought to the station on a standard-gage railway in 50-ton hopper-bottom cars from the Harwick Mine, which is

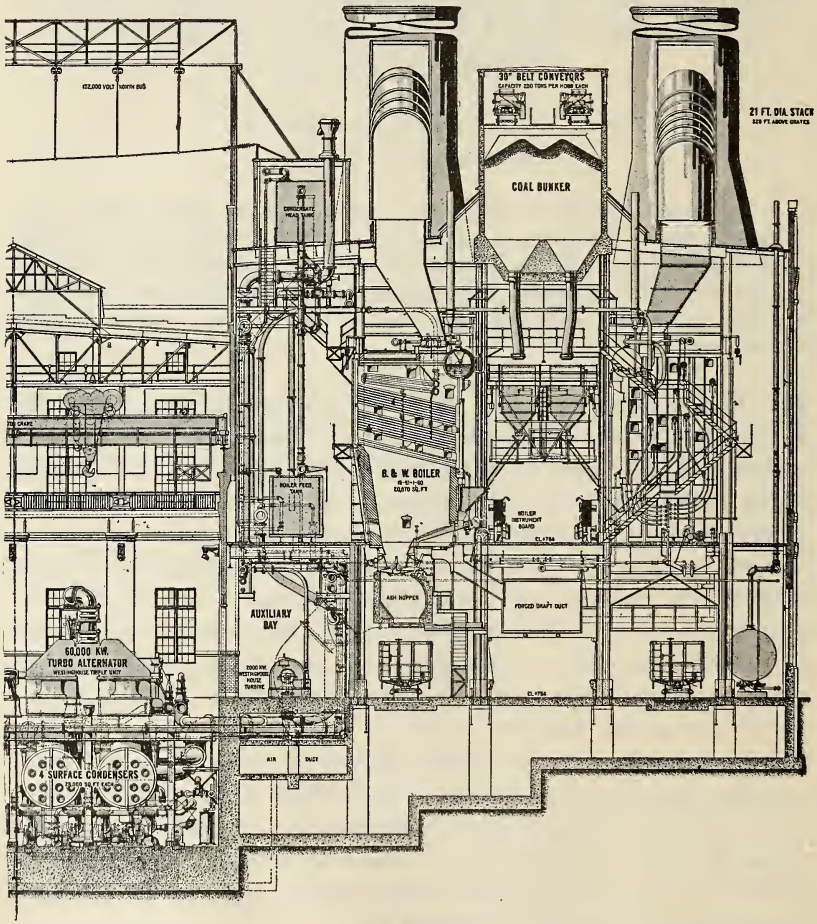


Fig. 1 Cross-Section of Boiler House and Part of Turbine Room

owned by the Duquesne Light Company and located about a mile and a half from the power station. A space about 1000 by 400 ft. just north of the power station is provided for coal storage and is large enough for the storing of about 150,000 tons of coal. A rope-

operated gantry bridge is now being erected for handling coal into and out of this storage. For the present coal is stored along the east side of the railroad trestle as shown in Fig. 3 and is handled by a 10-ton locomotive crane. As a measure against fire in this stored coal, temperature readings are taken at regular weekly intervals. Fig. 4 shows temperatures taken in the past.

4 At the plant coal is dumped into receiving hoppers under

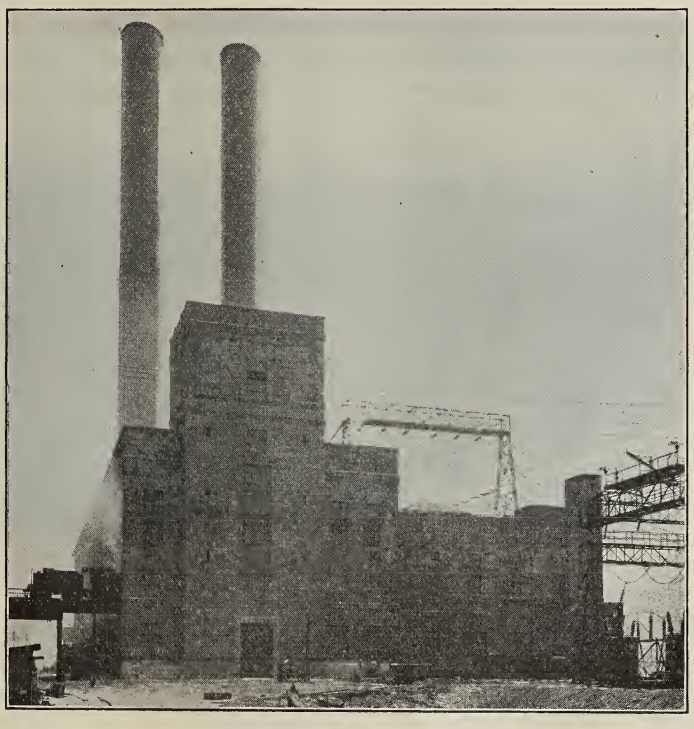


Fig. 2 General View of West End of Plant

the tracks. Crushers are located below the hoppers and the crushed coal is raised by two pivoted bucket elevators to distributing belt conveyors over the bunkers as shown in Fig. 5. As the coal goes to the crushers, water is added to help keep down the dust, and as an aid to combustion.

5 A representative sample of each day's coal is secured by taking individual samples of about 75 lb. from each carload as the cars are dumped. These are placed in airtight cans and so kept

until they are to be used. The whole day's samples are then run through an automatic crushing and sampling machine which extracts a 3 to 4 per cent sample, amounting at the present time to about 40 lb. This 40-lb. sample is quartered on a clean floor and two 3-lb. samples are taken out and put in sealed containers. One of these is sent to the laboratory for analysis and the other is held at the plant until satisfactory analysis of the laboratory sample has been completed. This extra sample is held so that, in case the first sample is lost or more is wanted for analysis, there will be some of the same lot available.

6 To secure samples of the coal as fired, a sample of about

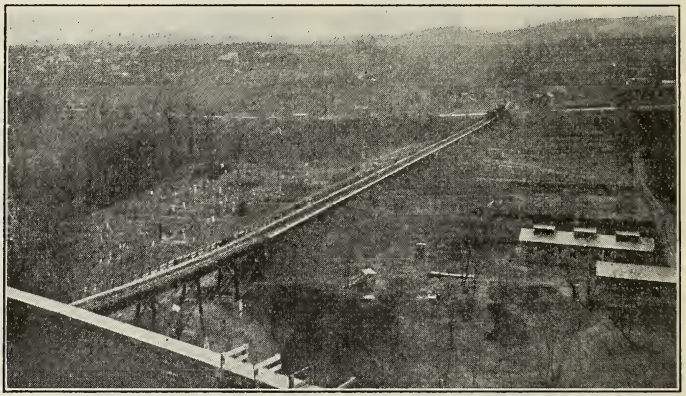


Fig. 3 General View of Coal-Storage Space

1000 lb. is also taken from the stoker hoppers at 7 a.m. daily. This sample is treated in the manner described above. Table 1 shows the average of a number of proximate analyses of the fuel and Table 2 a representative ultimate analysis.

7 Every precaution is taken to prevent loss of moisture from the samples before analysis. The methods for laboratory sampling and analysis of the American Society for Testing Materials are followed. An Emerson bomb calorimeter and thermometers, all calibrated by the Bureau of Standards, are used for determination of the heat value.

8 The samples taken from the cars as they are unloaded give advance information to the operators of variation in quality. The sample taken from the stoker hoppers is used as a basis for the daily-heat balance computations.

9 A representative ultimate analysis of the fuel used in the Colfax Station is given in Table 2.

10 In Table 3 are shown the fusion characteristics of the fuel.

TABLE 1 COAL ANALYSES

	As Received		Moisture-Free	
	As Unloaded	As Fired	As Unloaded	As Fired
No. of samples	25	30	25	30
Moisture, per cent	2.9	4.1		
Volatile matter, per cent	34.5	34.1	35.5	35.6
Fixed carbon, per cent	53.0	52.5	54.6	54.8
Ash, per cent	9.6	9.3	9.9	9.6
Total, per cent	100.0	100.0	100.0	100.0
Sulphur, per cent	1.0	1.1	1.1	1.2
Calorific value, B.t.u. per lb	13,351	13,237	13,758	13,895

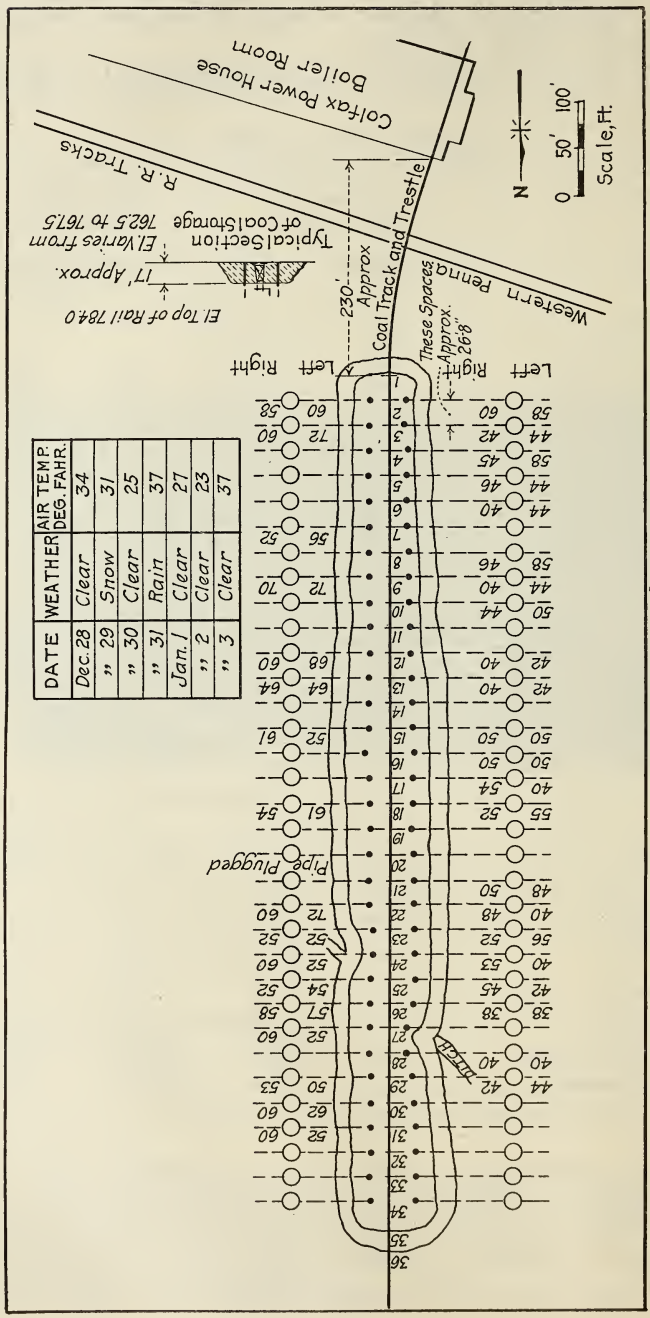
The difference in moisture between the coal as unloaded and as fired is due to the addition of water at the crushers mentioned above.

11 The coal bunker is of reinforced concrete and has a capacity of about 240 tons per boiler. Coal is not permitted to remain in the bunker longer than is required to maintain a reasonable reserve supply, which means a maximum of about three days. This

TABLE 2 REPRESENTATIVE ULTIMATE ANALYSIS OF FUEL

	As Received	Dry Coal	Moisture- and Ash-Free
Ash, per cent	8.88	9.30	.....
Sulphur, per cent	1.09	1.14	1.26
Carbon, per cent	72.92	76.40	84.23
Hydrogen, per cent	5.39	5.12	5.64
Nitrogen, per cent	1.49	1.56	1.72
Oxygen, per cent	10.23	6.48	7.15
Total, per cent	100.00	100.00	100.00
B.t.u. per lb. by calorimeter	13,202	13,832	15,251
B.t.u. per lb. by analysis	13,199	13,828	15,246

practically eliminates any possibility of spontaneous combustion in the bunkers. Distribution of the fuel to the indivdual stoker hoppers is by means of two weighing larries each having a capacity of 10 tons. A number of openings in the bottom of the bunker are provided with short cast-iron spouts equipped with chain-operated



DATE	WEATHER	AIR TEMP. DEG. FAHR.
Dec. 28	Clear	34
" 29	Snow	31
" 30	Clear	25
" 31	Rain	37
Jan. 1	Clear	27
" 2	Clear	23
" 3	Clear	37

FIG. 4 COAL-STORAGE SPACE, SHOWING LOCATIONS AT WHICH TEMPERATURE READINGS ARE TAKEN AND A SET OF READINGS TAKEN JAN. 3, 1922

gates and permit coal to be taken from any part of the bunker. Coal spouts with a 24-in. square opening are used.

12 After the larry weighing hopper is filled from a bunker chute, the operator balances the scales and prints the gross weight on a coal ticket by inserting the ticket into a slot on the beam weight and pressing a lever. After distributing the proper quantity of coal to a stoker, he prints the tare weight on the same ticket. He then subtracts the tare from the gross and enters the net weight of coal on his log sheet, charging it to the proper boiler and indicating the time. At the end of the day (12 midnight) the tickets, together with the log sheet, are collected and carried to the record

TABLE 3 SUMMARY OF FUSION TESTS OF COAL AS FIRED

Date of Sample	Per Cent Ash	Softening Temp., deg. fahr.	Softening Interval, deg. fahr.	Flowing Interval, deg. fahr.
July 19, 1921.....	9.96	2462	187	112
July 29, 1921.....	7.40	2426	71	90
August 5, 1921.....	9.96	2534	64	72
Sept. 1, 1921.....	10.32	2210	36	180

DEFINITIONS

Softening Temperature: The temperature at which the cone of ash has fused down to a spherical lump.

Softening Interval: The difference between the softening temperature and the temperature at which the first rounding or bending of the apex of the cone takes place.

Flowing Interval: The difference between the softening temperature and the temperature at which the ash has reached a fluid condition as shown by spreading out over the base in a flat layer.

clerk's desk. The following morning this clerk checks over the subtractions and adds up the total coal fed to each boiler for that day, these totals being posted to a boiler summary sheet for permanent record which is incorporated with the daily log sheets and filed.

STOKERS AND CLINKER GRINDERS

13 Each stoker is provided with an extension hopper, the capacity of the grate and hopper together being about 25,000 lb. The stokers are of the Westinghouse underfeed type and have 17 retorts of 20 tuyeres each. The projected grate area of each stoker is approximately 403 sq. ft.

14 Each stoker is driven by a 15-hp. variable-speed direct-current motor, equipped with both armature and field control. Current for these motors and for the clinker-grinder motors is supplied

by two 2200-kw. 440-volt a.c. to 125-volt d.c. motor-generator sets located in the auxiliary bay of the turbine room.

15 The stoker operator is guided entirely by observation of the fires and of the wind-box pressure. The forced-draft-fan speed and consequent wind-box pressure are automatically controlled by variations of the steam pressure in the main header. Wind-box pressure is for this reason somewhat of an indication of load condition. The pressure over the fire is maintained at from 0 to minus 0.1 in. of water by means of balanced-draft damper controllers.

16 In this type of stoker the upper and lower rams are linked

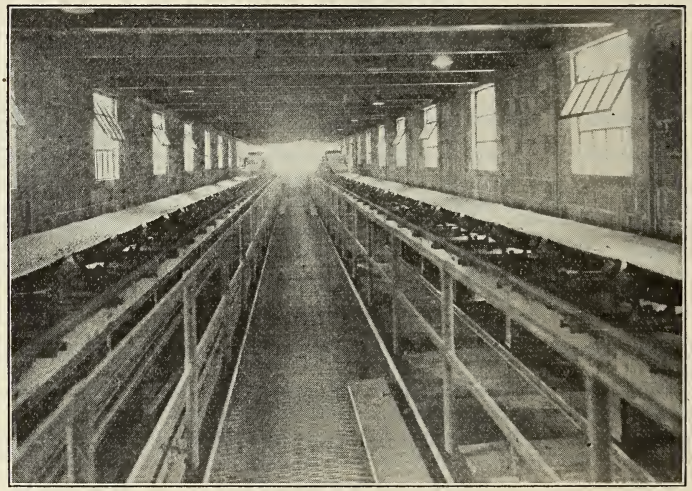


Fig. 5 View of Belt Conveyor

together, but the movement of the lower rams is made much less than that of the upper through lost motion in the linkage. In case the fire piles up on the lower end of the grate, this lost motion may be taken out by means of U-links which are left on until the fire is in proper condition. The stoker drive is capable of very close speed regulation and with ordinary care and the occasional use of the U-links, an even fuel bed suitable to the load may be maintained. Over the center of the firing aisle a master gage, load indicator and clock are located. The master gage indicates main-header steam pressure and has a 24-in. face. The main-unit load is indicated in thousands of kilowatts by means of changeable, illuminated numerals which are controlled by the switchboard operator

in the control room. Two more instruments are being added to this board, one to indicate the pressure in the forced-draft duct, the other to indicate the pressure in the boiler-feed header.

17 Each boiler is provided with an individual instrument and control board shown in Fig. 6. At the top are two pressure gages connected to either end of the boiler drum. Between them is a three-in-one draft gage which indicates the pressure in the wind box,

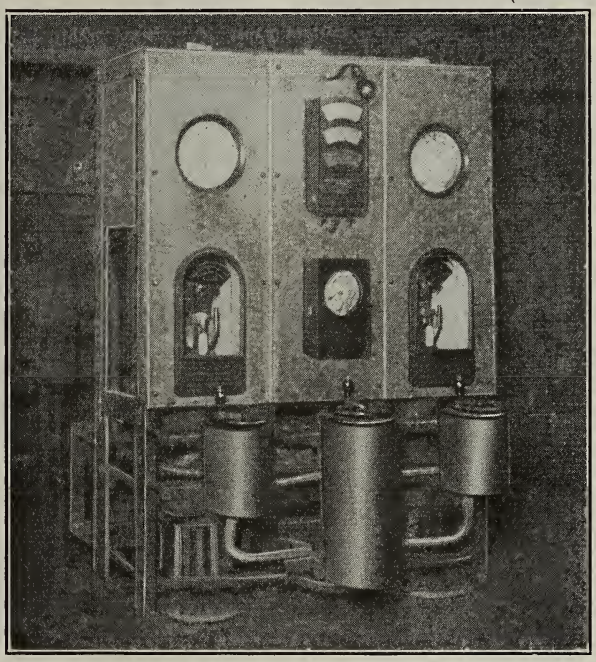


Fig. 6 Boiler-Instrument Board

over the fire and in the boiler uptake. Below the draft gage is a combined CO_2 and furnace-pressure recorder. Each of the two boiler-feed lines is provided with a Simplex venturi meter, the recorders for which are at the bottoms of the two side panels. Each of these meters has a capacity sufficient to feed the boiler at over 250 per cent of rating. The stoker and clinker-grinder motor controllers are bolted to the frame below the board, with the stoker-motor controller in the center and the clinker-grinder controllers at either side. This places practically the entire control for each boiler at one point so that but one stoker operator is required for

4175 nominal boiler horsepower — which is over 9000 developed boiler horsepower at 220 per cent rating, the normal daytime condition. No centralized control is provided as such a system is of doubtful advantage. The management of stokers, aside from that which can be made automatic, must be almost entirely through visual observation of the fuel bed. There are few factors of stoker

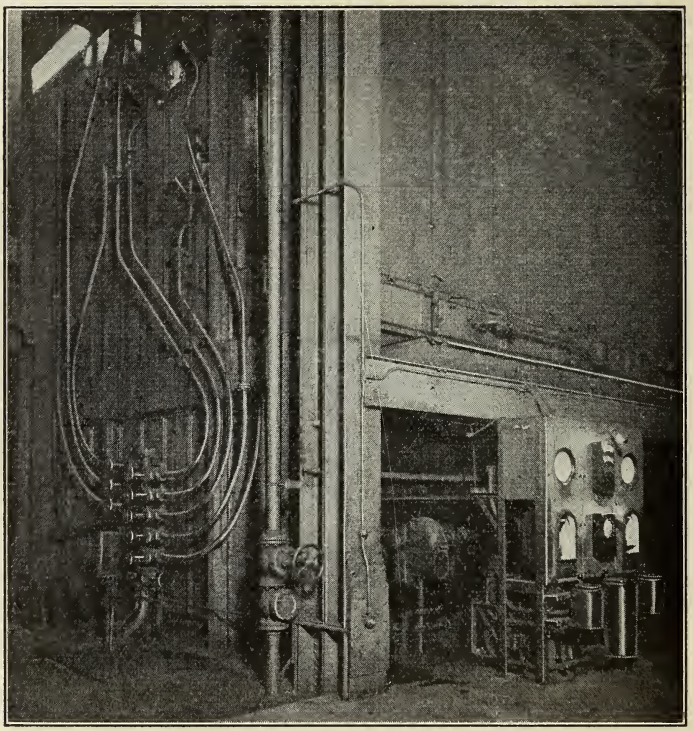


Fig. 7 General View of Boiler

control that can be centralized in such a way as to save either labor or fuel. Fig. 7 shows the general arrangement of the boiler, gage board, etc.

18 All superheater outlets are provided with a well for indicating and recording thermometers. Recording thermometers have not as yet been installed. Multiple-record pyrometers are used for recording the flue-gas temperatures.

19 The CO_2 recorders are checked frequently with an Orsat,

and have not been found satisfactorily accurate. This proved to be due largely to variations in the temperature of the water fed to them. Steps are being taken to provide water of a reasonably constant temperature which will eliminate this difficulty.

20 When this installation was first put in operation some difficulty was encountered with clinkering. After considerable experimenting it was definitely determined that this was due in part to periods when there was pressure over the fires and in part to improper water distribution in the clinker pits. Particular care is now taken to maintain a furnace pressure below atmosphere at all times, and the water distribution has been improved as described below with the result that clinker trouble has been practically eliminated.

21 Some trouble was at first experienced due to burning out of the lower front feed wedges. This trouble has been largely eliminated by allowing air to blow through from the wind box, thus keeping the wedges cool. To facilitate replacement, the air-box tops and grinder-pit aprons have been made sectional instead of as in the original design.

22 The stokers are provided with double-roll clinker grinders, divided in the center, with the rolls turning toward each other. The two halves are driven by separate motors, separately regulated, giving individual control over the two halves of the grinder. The rolls are driven through reciprocating pawls and ratchets which may be disengaged on either roll. It has not been found desirable to operate the rolls continuously. Under normal operating conditions the rear rolls are operated rarely; the front rolls are run for periods of from two to five minutes in every twenty minutes, this meaning from six to eight revolutions in one hour. Grinder operation is governed entirely by observation of the ashpit, the top of the ash bed is kept above the top of the rear-wall air boxes.

23 To prevent the formation of clinker masses in the ashpit and to cool the clinker rolls, water is introduced above the rolls. At first this was done through cast-iron spray heads about eighteen inches high, which were mounted one on each of the five grinder-roll bearings. These heads sprayed water horizontally in all directions from their tops. There were also horizontal spray pipes, located below and parallel to the clinker rolls, which sprayed water up against the rolls for cooling purposes. It was found that though this served as a very satisfactory cooling arrangement, the spray

heads had a tendency to hold up the clinker masses and prevent their entering the grinders. Low, wedge-shaped bearing caps which sprayed water horizontally were substituted for the spray heads. These caps did not hold up the clinker and it could be ground out, but the grinding out of large clinker masses placed severe duty on the grinders. It was felt that this duty could at least be lessened if the formation of hard clinker could be decreased. Neither of the above arrangements put water high enough in the ashpit to prevent the formation of hard clinker. Finally, new upper-grate-box tops having a down-turned projecting lip were installed. Under this lip was placed a horizontal spray pipe perforated with quarter-inch holes six inches apart. This pipe introduces the water high up in the ashpit and quite effectually prevents the formation of hard clinker masses. The use of all the other spraying methods was discontinued. The introduction of water high up in the ashpit tends to quench the ash before combustion is as complete as could be desired thus increasing the amount of combustible in the ash. At the present time this seems to be the lesser of two evils. The sprinkling system requires about fifty gallons per minute for each boiler.

24 At present the ashpit water runs out around the edges of the ash gates to the boiler-room basement floor, where it is carried away by a system of large drain pipes. A water-catching system attached to the ash gates will be installed with the next boiler installation and if it proves successful all of the gates will be so equipped.

ASH HANDLING AND SAMPLING

25 The clinker grinders discharge into firebrick-lined reinforced-concrete ash hoppers with an approximate capacity of 60 cu. yd. These hoppers will hold the refuse of from one and a half to two days' normal operation. It is the regular practice, however, to remove the refuse from all working boilers twice a day. Each hopper is equipped with three 5-ft. square, air-operated, flat, sliding gates. Standard-gage railway tracks are provided and there is sufficient clearance so that standard railway cars may be run under the hoppers for ash removal.

26 The brick lining of the ash hoppers is standing up very well after eighteen months of operation, no repairs having as yet been required.

27 A sample of ash from each boiler is taken every other day.

A gross quantity of about 400 lb. is taken from the total 24-hr. dump from each boiler, the sample being taken from six different points in the pile, at a depth of about three feet below the surface

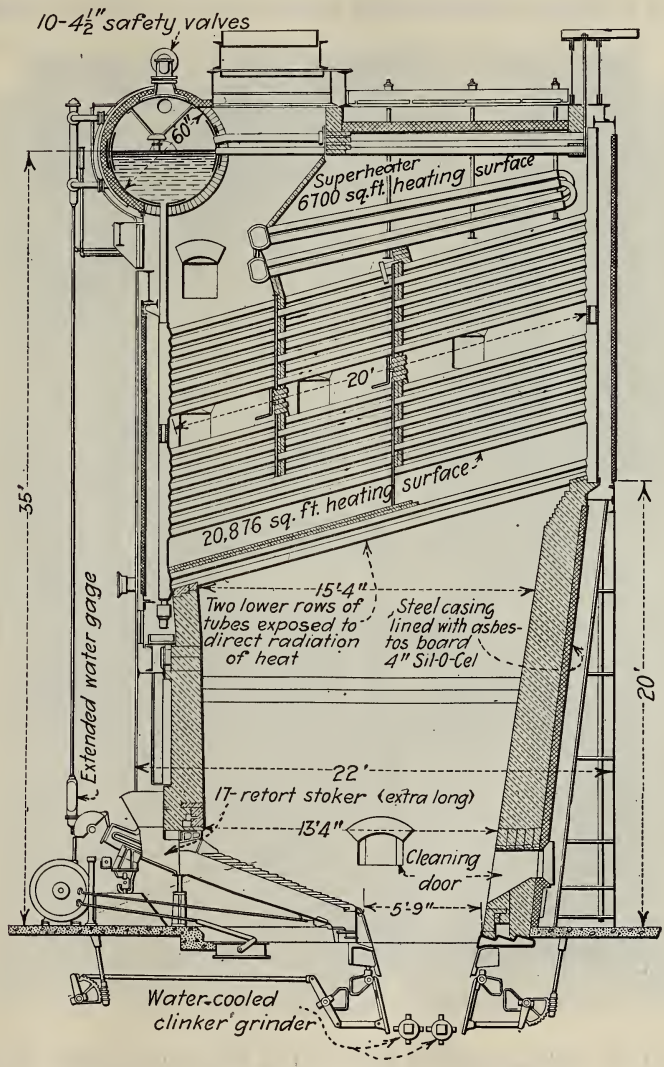


Fig. 8 Cross-Section of Boiler and Setting

of the pile. The whole sample is then hand-crushed and quartered until about three pounds remain, which is sealed in a container and sent to the laboratory to be analyzed for combustible. The method

of analysis is in accordance with the specifications of the American Society for Testing Materials.

28 Two laborers are required for ash handling. Part of the time of a dinky locomotive and operator is chargeable against ash

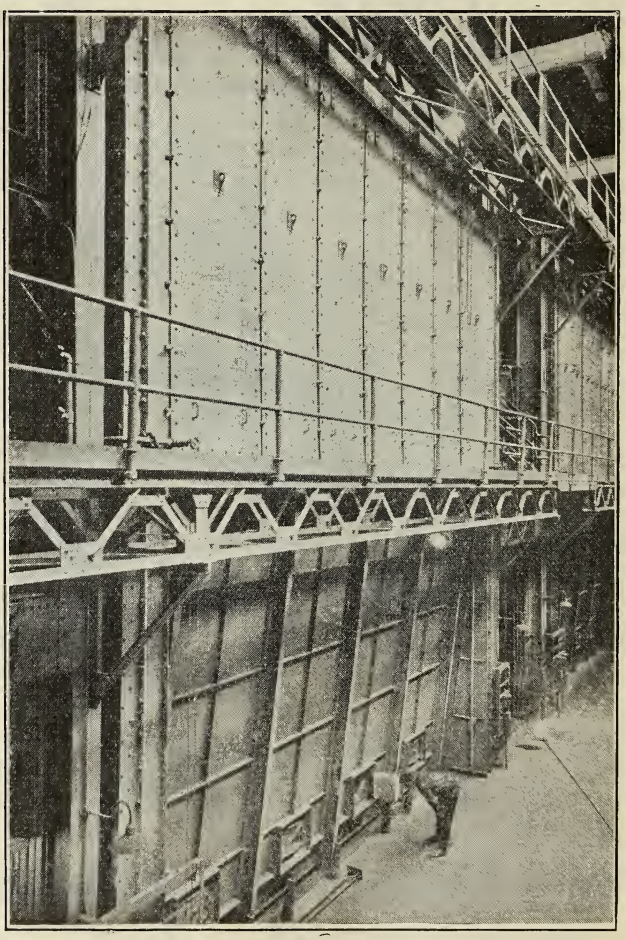


Fig. 9 Rear of Boiler

removal, as well as part of that of the locomotive-crane operator. The cars are moved by dinky locomotive and ashes are spread where desired by the locomotive crane. The necessity for the crane is only temporary while ashes are being used for grading. Ashes are also being removed in part by motor truck, and if desired can

be efficiently so handled even when the station is operating up to a 120,000-kw. load.

BOILERS

29 There are at present seven boilers of the Babcock & Wilcox cross-drum type, 18 tubes high by 51 wide, with 20,876 sq. ft. of

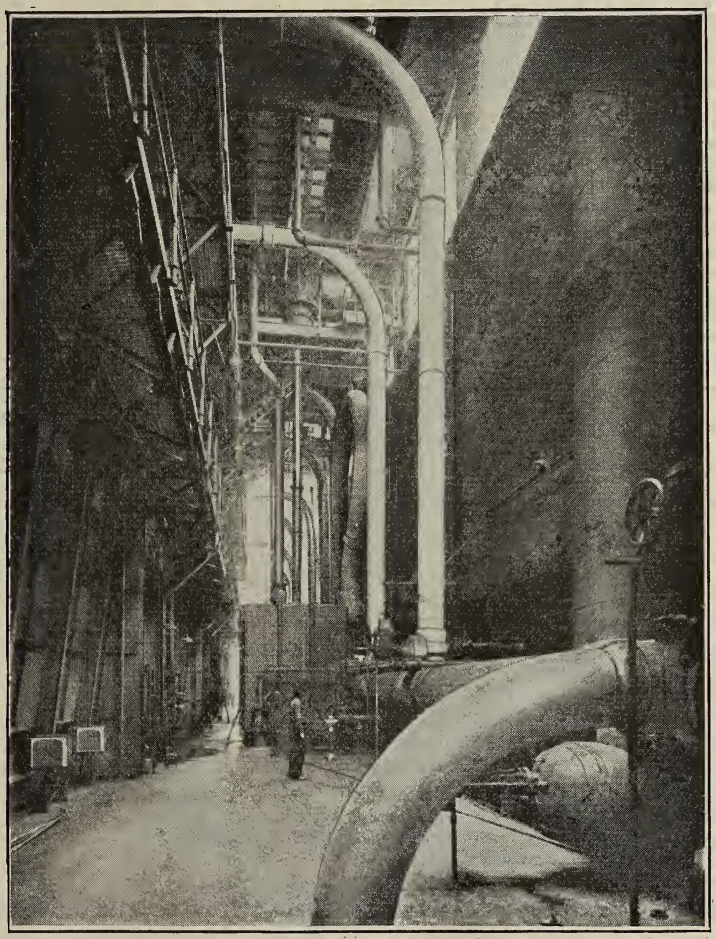


Fig. 10 Rear of Boiler

heating surface and with the so-called "Alert" baffling. There are 918 tubes in each boiler proper and 102 circulating tubes connecting the uptake headers with the drum, making 1020 tubes in all.

Fig. 8 is a cross-section of the boiler and setting. It will be noted that the combustion space is large. The present settings contain 0.345 cu. ft. of combustion space per square foot of heating surface and those to be installed this summer will be the same. Figs. 9 and 10 show the rear of the boilers, boiler-feed tank and other apparatus.

30 Each boiler has two water columns with high- and low-

TABLE 4 TEMPERATURE TRAVERSES IN BOILER NO. 5 UPTAKE

Test number	1		2		3			
Date of test	2–4–22		2–13–22		2–13–22			
Starting time	8:55 A.M.		10:13 A.M.		1:15 P.M.			
Duration of test, hr.	2		1½		1½			
Number of positions	12		7		7			
Number of readings in each position	4		3		3			
Interval between readings, min.	3		3		3			
Average values								
Boiler rating, per cent	224		211		206			
CO_2 by Hays recorders	11.8		12.0		13.1			
Flue-Gas Temperatures	Fixed [2]	Movable [3]	Fixed [2]	Movable [3]	Fixed [2]	Movable [3]	Fixed [2]	Movable [3]
Position [1] No. 1	483	439	480	435	473	428	479	434
2	489	462	...	...	...	...	...	...
3	481	479	477	483	473	489	477	484
4	477	481	...	...	...	...	...	...
5	476	481	485	499	475	491	479	490
6	474	477	487	500	482	490	481	489
7	477	478	...	...	...	...	...	...
8	482	482	488	490	480	480	483	484
9	486	486	...	...	...	...	...	...
10	482	472	485	480	480	471	482	474
11	484	471	...	...	...	...	...	...
12	484	460	484	461	485	462	484	461

[1] Position numbers start from rear of boiler, read to front. Length of traverse, 65 in. Place of measurement, below damper neck in center of boiler.

[2] Fixed instrument was a mercury thermometer (25 in. in uptake)

[3] Movable instrument was a Foxboro recorder.

water alarm whistles. When one of these alarms sounds the operator examines the feed regulators for possible sticking, and if necessary cuts it out and uses hand control until repairs can be made. If the regulator is found to be all right he checks up the feed pressure and if necessary starts another pump or puts on the auxiliary feed. Extended water columns, which can be read at the operating floor level, were installed but due to improper design could not be de-

pended upon. These columns have been modified and are being installed again.

31 Furnace temperatures vary from 2500 to 2800 deg. fahr.;

TABLE 5 CO_2 TRAVERSE OVER FIRE IN BOILER NO. 5

Number of Test	1		2		3		4		Average	
Date of test	2–11–22		2–11–22		2–17–22		2–17–22			
Starting time	10:55 A.M.		11:35 A.M.		10:10 A.M.		11:15 A.M.			
Duration of test, hr.	½		½		½		½			
Number of positions	7		7		7		7			
Readings in each position	1		1		1		1			
Interval between readings, min	5		5		5		5			
Average values										
Boiler rating, per cent	213		189		196		201		200	
Wind-box pressure, in. of water	3.7		3.6		3.8		3.9		3.7	
Over-fire draft, in.	0.06		0.04		0.02		0.0		0.03	
Before-damper draft, in.	0.88		0.75		0.76		0.81		0.80	
Steam pressure, lb. per sq. in.	275		275		274		273		274	
Steam temperature, deg. fahr.	548		545		548		554		549	
Stack temperature, deg. fahr.	483		475		476		480		479	
CO_2 over fire, per cent	14.0		11.4		13.0		11.0		12.3	
CO_2 3rd pass stationary, per cent	13.5		12.2		12.9		11.5		12.5	
CO_2 by Hays recorder, per cent	12.5		11.5		12.8		13.0		12.5	
	Stat. 3rd Pass	Trav. over fire	Stat. 3rd Pass	Trav. over fire	Stat. 3rd Pass	Trav. over fire	Stat. 3rd Pass	Trav. over fire	Stat. 3rd Pass	Trav. over fire
CO_2 1st position	11.8	12.2	10.4	7.0	11.6	8.5	11.6	9.1	11.4	9.2
2nd	13.3	13.9	10.7	8.8	11.5	10.2	12.0	8.6	11.9	10.1
3rd	14.3	15.5	11.7	10.5	13.4	13.8	11.1	9.0	12.6	12.2
4th	14.0	14.6	12.1	12.2	13.8	14.3	11.6	10.6	12.9	12.9
5th	13.5	14.7	12.6	12.5	13.7	14.7	11.4	12.8	12.8	13.7
6th	13.8	14.3	14.2	14.5	13.5	14.7	11.3	13.2	13.2	14.2
7th	14.0	14.0	14.0	14.5	12.8	15.0	11.3	13.9	13.0	14.4

Position numbers start from rear of boiler, read toward front.
Length of traverse over fire equals 76 in.
Place of measurement 52 in. below tube header center of boiler.
Length of stationary instrument 35 in. in pass, top of 3rd pass.

at the top of the first pass this approximates 1000 deg. fahr., between the second and third passes it is 530 deg. fahr., and the exit gas temperature varies from 450 to 480, usually being about

470 deg. fahr. Fig. 11 is a chart showing the temperature range and gas velocities through the boiler. Typical temperature and CO_2 traverses are shown in Tables 4 and 5.

32 Slagging on the boiler tubes is slight and has never been sufficient to cause trouble. In one boiler taken off the line recently, 75 per cent of the first-pass area was stopped with slag, due to trouble in the stoker-ram throats. This is the only instance of serious slagging that has occurred. The use of distilled make-up water eliminates trouble from scale and consequent tube renewals. The total commercial operation of the seven boilers now installed has been approximately 39,922 boiler-hours up to February 1, an

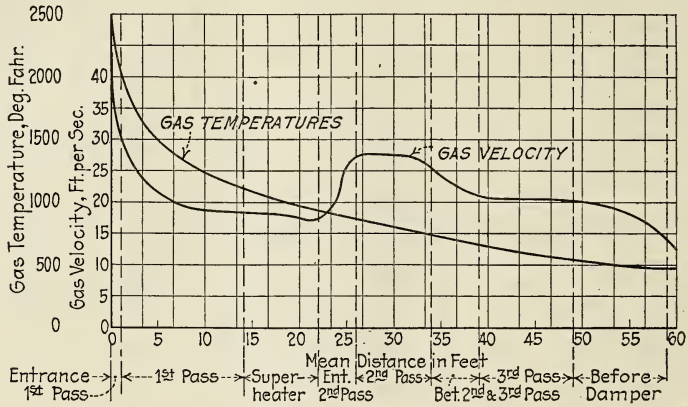


Fig. 11 Variation of Temperature and Velocity of Flue Gas Throughout the Boiler

average of 5703 hr. per boiler, and but two tubes have been replaced.

33 Each boiler is equipped with eighteen soot-blower elements, nine on each side. Trouble was experienced from warping of the elements apparently due to insufficient drainage. The drip system was altered so as to carry this water off more positively, and this sort of trouble was eliminated. Future installations will be modified so that no valves will be required on the individual soot blower leads. This will permit easy draining of accumulated water and will require but a single master valve on each side of the boiler. Soot blowers are operated three times a day with the dampers opened wide.

34 The regular procedure for taking a boiler off the line is as follows: When the boiler and setting are cool enough the water is

drawn out out of the boiler through the blow-off lines. A large tank is provided into which the water from the boilers may be run, and, after settling, pumped to the main storage tank, or returned to the boiler. The present practice is to waste the water to the river when emptying boilers. It has been found unnecessary to blow down at all between cleanings, and wasting this water is a precaution against the accumulation of impurities in the system. If a boiler has to be emptied after a short run the water is returned to the system by means of the tank and pumps mentioned above.

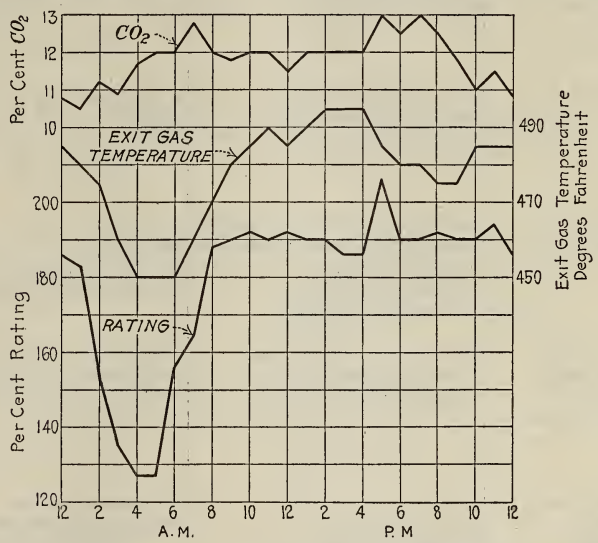


Fig. 12 Hourly Variation in Rating, Gas Temperature and CO_2 for Typical 24-Hr. Day

When the boiler is empty, all valves are locked and posted with danger signs.

35 The furnace cleaners first remove any accumulation of slag or soot from the tubes, passes and walls. The clinker-grinder pit is then emptied, inspected and repaired. After this, refuse remaining on the grate is moved into the grinder pit and left there to act as a seal when starting a new fire. The stoker, stoker drive and brickwork are then inspected and repaired. A certain number of tube caps are then removed, and the tubes inspected. A system is followed in tube inspection so that a new lot of tubes are covered each time a boiler is open. Tube caps are replaced and made tight,

TABLE 6 PIPING SCHEDULE OF

Service	Pipe	Fittings
High pressure Main and auxiliary Superheated steam High-pressure drip system	2″ and under E.S. steel 2½″ to 12″ F.W. steel 14″ and over O.D. steel ⅜″ thick	1½″ and under E.H. scr. F.S. 2″–12″ E.H. flgd. C.S. 14″ and over E.H. flgd. C.S. (special ports)
H.P. saturated steam	2″ and under E.S. steel 2½″–12″ F.W. steel	2″ and under E.H. scr. C.S. 2½″–12″ E.H. flgd. C.S.
Feedwater lines	3″ and under I.P.S. brass 3½″ and over F.W. steel	2″ and under E.H. scr. C.I. 2½″ and over E.H. flgd. cast semi-steel
Boiler blow-off Superheater blow-off	Boiler side B.O. Val. E.S. steel 2″ and under E.S. steel. 2½″ and over (Inside Station). F.W. steel, (Outside sta.) E.H. flgd. cast semi-steel	Boiler side B.O. valve E.H.flgd. C.S. 2″ and under E.S. scr. C.I. 2″ and over E.H. flgd. C.I.
Feed-suction piping Low-service piping Hot-well piping Circ. and misc. water	2″ and under E.S. galv. steel 2½″–3″ std. galv. steel 4″ and over F.W. steel or C.I.	3½ and under std. scr. C.I. 4″ and over std. flgd. C.I.
Main and aux. exhaust Vacuum lines Compressed air Low-pressure drips Misc. L.P. steam	2″ and under E.S. steel 2½″–12″ std. steel 14″–22″ O.D. steel 5/16″ thick. 24″ and over Std. C.I. for horiz. runs, O.D. steel 5/16″ thick for vertical run	3½″ and under std. scr. C.I 4″ and over std. flgd. C.I
Free exhaust	12″ and under std. steel 14″–22″ O.D. steel 5⅛″ thick 24″ and over std. C.I. vert. spiral riveted gal. steel	Std. flgd. C.I.
Oil	2″ and under E.S. steel 2½″ and over std. steel	3½″ and under std. scr. C.I.

E.S. = Extra Strong, E.H. = Extra Heavy, C.I. = Cast Iron, F.S. = Forged Steel, I.P.S. = Iron Pipe Size, F.W. = Full Weight. Scr. = screwed. All semi-steel material to be tested to

COLFAX STATION

Valves	Gaskets	Flanges	Bolts
1″ and under E.H., F. S. and monel threaded 1¼″–1½″ E.H., C.S. and monel O.S. and Y threaded 2″ and over E.H. flgd. C.S. and monel O.S. and Y.	Compressed asbestos 1/16″ thick, such as "Durabla"	2″–3½″ E.H.F.S. welded to pipe 4″ and over E.H.F.S. Van Stone	Special oil-heat-treated with nuts on each end
2″ and under E.H. scr. C.S. monel mounted 2½″ and over E.H., C.S. flgd. monel mounted (O.S. and Y. pattern)	Compressed asbestos 1/16″ thick	2½″–5″ E.H.F S. welded to pipe 6″ and over E.H.F.S. Van Stone	Special oil-heat-treated with nuts on each end
2″ and under E.H. scr. all bronze 2½″ and over E.H. flgd. semi steel and bronze (O.S. and V pattern)	Compressed asbestos 1/16″ thick	3″ and under E.H.F.S. screwed 3½″ E.H.F.S. welded; 4″ and over E.H.F.S. Van Stone	Std. steel bolts
2″ and under E.H. scr. C.I. and bronze 2½″ and over E.H. flgd. C.I. and bronze O.S. and Y	Compressed asbestos 1/16″ thick	Boiler side B.O. valve E.H.F.S. welded E.H. flgd. C.I. scr. for steel pipe Cast integral for cast semi-steel pipe	Std. steel bolts
2″ and under std. scr. composition (inside screw pattern) 2½″–3½″ std. C.I. and bronze, scr. 4″ and over std. flgd. C.I. and bronze (O.S. and Y. pattern)	Rainbow or black ebonite 1/16″ thick or canvas and red lead	Std. C.I. threaded for 4″ to 12″ steel pipe 14″ and over std. C.I. Van Stone cast integral for cast-iron pipe	Std. steel bolts
2″ and under std. scr. composition (inside screw pattern) 2½″–3½″ std. scr. C.I. and bronze. 4″ and over std. flgd. C.I. and bronze. (O.S. and Y pattern)	Rainbow 1/16″ thick	12″ and under std. C.I. scr. 14″–22″ std. C.I. Van Stone 24″ and over cast integral for C.I., std. C.I.V.S. for steel	Std. steel bolts
Relief valves C.I. and bronze water sealed 18″ and over with hydraulic cylinders and remote control	Rainbow 1/16″ thick	12″ and under std. C.I. 14″–22″ std. Van Stone 24″ and over cast integral vert. spiral riv. std. C.I.	Std. steel bolts
2½″ and under std. scr. composition 4″ and over std. flgd. C.I. and brass (inside screw pattern)	Compressed asbestos	4″ and over std. C.I. companion	Std. steel bolts

600 lb. All valves provided with removable seats and disks.

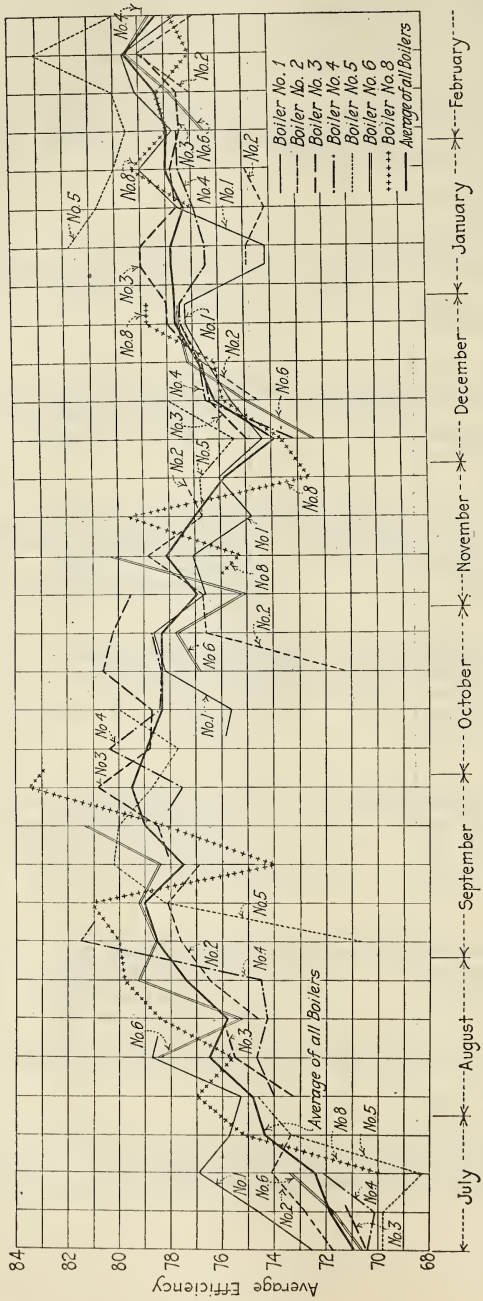


FIG. 13 AVERAGE WEEKLY EFFICIENCIES OF INDIVIDUAL BOILERS

FIG. 14 AVERAGE WEEKLY BOILER PERFORMANCE FOR ALL BOILERS

Note—The points on this curve are the weighted averages of the average weekly efficiencies for the individual boilers, hence the variations at some points.

after which a thorough final inspection of the entire boiler is made by the boiler engineer. The boiler is then filled full from a special boiler refill line which takes water from the main storage tank or from the emptying tank mentioned above, A hydrostatic test is then made to locate leaky caps, etc. If everything is found to be in proper order the water is let out down to the normal working level and the boiler is ready for operation. The work of inspection and repair outlined above ordinarily takes from ten to fourteen days.

36 The stoker hopper is then filled and the stoker grate cov-

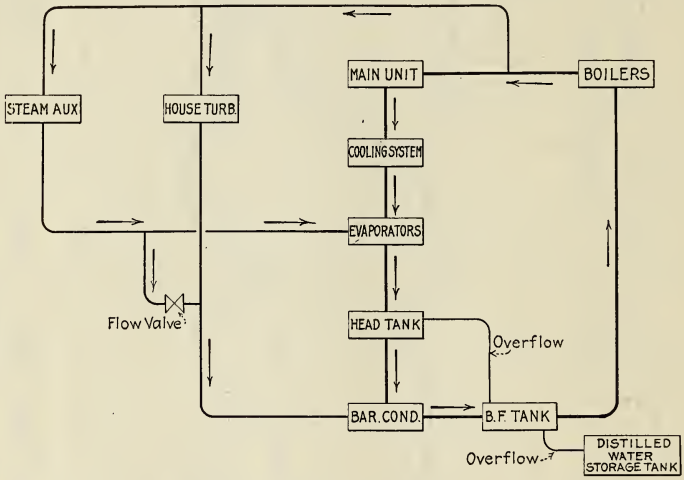


Fig. 15 Diagram of Steam and Feedwater Cycle

ered nearly to the edge of the grinder pit. Wood is piled on top of the coal, soaked with kerosene and the fire is lighted. Ordinarily a slow fire is maintained and the boiler heated up gradually, so that twelve to fourteen hours are required before the boiler can be put on the line. In an emergency a boiler can be put on three hours after the fire is started. Full efficiency is not reached until from four to seven days after the boiler is put on the line. In the past boilers have been taken off for cleaning and repairs after 30 to 45 days of service, but it is expected that in the future service periods will be from 90 to 100 days. This increase is due to some slight modification of the settings and increased knowledge of the performance of the equipment.

37 The settings are of firebrick throughout, the brick walls

being 18 in. thick. Sil-O-Cel and asbestos millboard fill the $4\frac{1}{4}$ in. inch space between the firebrick and the steel casing. Deterioration of the setting is very slight.

38 The formation of clinker on the furnace walls is prevented by air cooling, Bernitz and Drake blocks being used. These blocks did not altogether prevent clinkering on the side walls and side-wall tuyeres have been installed. These have practically eliminated all clinkering on the side walls. Such clinker as does form is easily removed. Special tuyere-type, ram-box tops which

Fig. 16 Comparison of V-Notch and Venturi Records

blow air in directly over the retorts prevent clinkering and scoring of the front wall.

39 The superheaters are located over the tube banks between the first and second pass. Considerable trouble has been experienced because of insufficient superheat. The normal superheat should be 180 deg., but 145 deg. has seldom been exceeded. The boilers to be installed this summer will have the superheaters located above the sixth row of tubes, and with this arrangement no trouble from deficient superheat is anticipated.

40 There is one forced-draft fan for every four boilers. The fans are of the radial-flow type, and have a capacity of 250,000 cu. ft. of free air per minute at a maximum static pressure of 6 in. of water. The fans are driven through reduction gears by 420-hp. turbines. Each fan turbine is provided with a speed controller actuated by variations of steam pressure in the main header.

Ordinarily no control of the stoker air other than that provided by the automatic forced-draft-fan speed control is used, although individual wind-box dampers are provided that may be used in case of emergency. There is one main forced-draft duct, 9 ft. 6 in. by 15 ft., running beneath the firing-aisle floor, from which all stokers are supplied. This duct is provided with sectionalizing dampers located midway between the fan inlets. At each boiler a branch duct runs from the main duct to the stoker wind boxes. The duct system is of $\frac{3}{16}$-in. sheet steel made practically airtight without welding.

41 The generator-cooling-air discharge is carried to the main

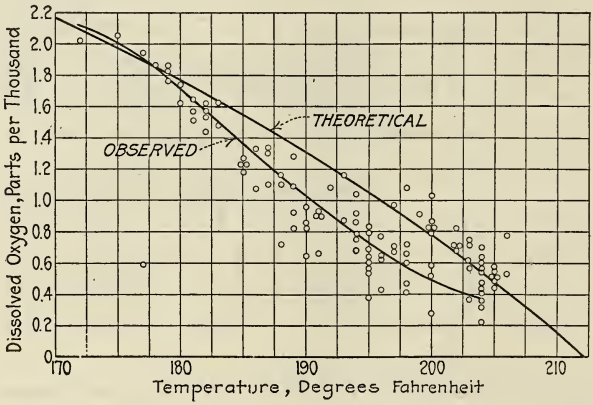


Fig. 17 Comparison of the Actual Dissolved Oxygen in Colfax Boiler Feedwater with Theoretical Solubility at Varying Temperatures

air-supply duct from which the fans take part of their supply. In addition to this, large openings are provided at the tops of the fan chambers through which air may be drawn from the top of the auxiliary bay. In winter the openings between the fan chambers and the boiler-room basement are closed. This is done to prevent freezing of piping due to drawing large quantities of air from outdoors through the boiler-room basement.

42 Fig. 12 shows the hourly variation in boiler-room operation over a typical 24-hr. period. Fig. 13 shows the average efficiency, rating and gas temperature for all boilers for the period from July, 1921, to February, 1922. The low point at the beginning of December was due to a period during which there was considerable trouble with clinker.

PIPING SYSTEM

43 The piping system for boiler feeding consists of a main and auxiliary header. The main header runs below the firing-aisle floor and between it and the forced-draft duct. A feed line is carried up each side of the boiler (these boilers are provided with feed connections at both ends of the drums) and is controlled by globe valves at the boiler-room floor level. Automatic boiler-feed regulators are provided on the feed lines at the boiler drums, part being Stets and part Copes regulators. The auxiliary header runs over and at the rear of the boilers. The auxiliary feed line on one side of each boiler is provided with a by-pass through the automatic controller, that on the other side being provided with hand control only. The boiler-feed system in general is described below.

44 The piping system was installed in accordance with the general piping specification, Table 6.

45 Fig. 14 shows the individual efficiency for all boilers from July, 1921, to February, 1922. This clearly shows the steady increase in efficiency as conditions were improved and as the operating force became more familiar with the equipment. The low point in December is explained above.

46 No serious trouble with piping or valves has been experienced. One fitting on the boiler-feed system cracked, but it was found upon checking up with the manufacturer that that particular fitting was gray cast-iron instead of semi-steel as specified. Some trouble was also encountered with leaky steel globe valves. These valves were found to be defective due to " wartime workmanship " and were replaced by the manufacturer. All bronze safety-valve feathers had to be replaced with monel metal, as bronze was found to be affected by the temperature. Bronze-mounted gage-glass cocks had to be replaced with monel-mounted cocks, as the former would not stand up under the service.

FEEDWATER SYSTEM

47 The feedwater cycle is shown diagramatically in Fig. 15. All water in the system is distilled. Make-up water is provided by two evaporators of the double-effect, dry-tube type. Water from the evaporators is pumped directly to the head tanks, above the barometric condensers, which serve as feedwater heaters. The

average make-up required is about 1.9 per cent of the total amount flowing in the system. The tail pipe of the barometric condenser

TABLE 7 TYPICAL WATER ANALYSES

Source	Allegheny River	Condensate Hotwell	Distillate No.1 Unit	Boiler-Feed Tank	Storage Tank	Colfax Well
Date Sampled	9-14-21	10-21-21	10-5-21	11-17-	10-10-21	10-14-21
	Values expressed in parts per million.					
Total Solids (evap.)	182.4	3.45	4.43	20.43	15.60	321.2
Incrusting Solids	125.6	2.77	3.42	17.02	11.65	299.5
Non-Incrusting Solids	47.1	8.61	11.16	10.70	10.42	92.9
Suspended Matter	35.5	0.00	0.00	0.00	0.00	6.63
Alkalinity as $CaCO_3$	14.5	3.86	3.97	3.97	8.76	146.8
Free Carbonic Acid	5.0	1.04	1.56	1.23	0.00	15.4
Probable Combinations (calculated)						
Ignition Loss	13.2	0.00	0.40	0.88	1.44	3.8
Silica	4.0	1.18	0.95	13.35	2.12	9.8
Oxides Iron and Alum	1.4	0.18	1.13	0.48	1.32	0.6
Calcium Carbonate	0.0	0.00	0.00	0.00	3.20	0.0
Calcium Bicarbonate	23.4	0.97	1.05	2.75	3.40	233.6
Calcium Sulphate	67.3	0.00	0.00	0.00	0.00	3.9
Magnesium Carbonate	0.0	0.00	0.00	0.00	0.00	0.0
Magnesium Bicarbonate	0.0	0.44	0.29	0.44	1.61	0.0
Magnesium Sulphate	29.5	0.00	0.00	0.00	0.00	51:6
Magnesium Chloride	0.0	0.00	0.00	0.00	0.00	0.0
Magnesium Nitrate	0.0	0.00	0.00	0.00	0.00	0,0
Sodium Bicarbonate	0.0	4.96	5.21	3.28	4.03	0.0
Sodium Sulphate	15.5	0.64	2.27	2.77	2.77	42.0
Sodium Chloride	30.7	2.75	3.51	4.56	3.45	47.3
Sodium Nitrate	1.0	0.26	0.17	0.09	0.17	3.7
Total	185.9	11.38	14.98	28.60	23.51	396.2
Cations, dissolved, determined						
Calcium (Ca)	25.6	0.25	0.25	0.68	2.12	57.8
Magnesium (Mg)	6.0	0.07	0.05	0.07	0.27	11.1
Sodium (Na)	17.3	2.71	3.59	3.61	3.40	33.2
Anions, dissolved, determined						
Bicarbonate (HCO_3)	17.6	4.71	4.84	4.84	6.81	179.1
Sulphate (SO_4)	81.6	0.45	1.56	1.89	1.89	69.6
Chloride (Cl)	18.6	1.65	2.12	2.75	2.10	28.7
Nitrate (NO_3)	0.7	0.18	0.09	0.07	0.09	2.7
Carbonate (CO_3)	0.0	0.00	0.00	0.00	1.91	0.0

is sealed in the boiler-feed tank from which the feed pumps take their suction. Two V-notch meters are provided in this tank; one

measures the water going to the feed pump and the other the overflow. All overflow from the system is piped to the boiler-feed tank so that it is recorded by this meter. Overflow from the boiler-feed tank flows to the main storage tank mentioned below. There is some variation between the water quantities recorded by the venturi and V-notch meters. Fig. 16 has been plotted showing these variations as percentages of the quantities recorded by the venturi

TABLE 8 OBSERVATION OF DISSOLVED OXYGEN IN BOILER FEEDWATER, PARTS PER 1000 BY VOLUME

Head Tank			Boiler-Feed Tank		
Temp., Deg. Fahr.	Actual Dissolved Oxygen	Theoretical Dissolved Oxygen	Temp. Deg. Fahr.	Actual Dissolved	Theoretical Dissolved Oxygen
100	(6) 1.93	4.7	(1) 185	1.21	1.55
101	(10) 2.10	4.65	(2) 186	1.2	1.50
102	(4) 2.15	4.6	(1) 187	1.34	1.45
103	(2) 2.05	4.55	(1) 188	1.11	1.40
104	(5) 2.43	4.5	(1) 189	0.81	1.35
105	No data	4.45	(3) 190	0.88	1.30
106	(3) 1.94	4.4	(2) 191	0.895	1.25
107	(1) 1.99	4.35	(1) 192	1.08	1.20
Mean 102	(32) 2.11	4.6	(0) 193		1.15
			(5) 194	0.75	1.00
			(8) 195	0.65	1.06
			(3) 196	0.70	0.95
			(0) 197		0.90
			(3) 198	0.506	0.85
			(0) 199		0.80
			(2) 200	0.43	0.75
			(0) 201		0.70
			(0) 202		0.60
			(0) 203		0.55
			(1) 204	0.21	0.50
			Mean 193	(34) 0.766	1.15

Figures in parentheses indicate number of observations at given temperature.

meters. It is the venturi-meter readings which have been used as a basis for all results given in this paper.

48 There are three 4-stage, 1500-g.p.m., turbine-driven centrifugal, boiler-feed pumps, regulated to maintain a pressure in the feed system of 350 lb. per sq. in. One pump is sufficient to supply the total feedwater for a load of about 60,000 kw.

49 Condensate from the main unit goes first to a surface heater where its temperature is raised about 4.5 deg. fahr. by the transformer- and turbine-oil cooling water. It then goes to the

evaporator condenser serving there as a circulating medium, and its temperature is increased about 32 deg. fahr. From the evaporator condenser it goes to the head tank and then through the barometric condenser. The exhaust from the house turbine and that part of the exhaust from the auxiliaries which is not used in the evaporators is here condensed, bringing the temperature of the mixture to about 205 deg. fahr.

50 The storage tank shown in the diagram has a capacity of

TABLE 9 OPERATING FORCE FOR THREE EIGHT-HOUR SHIFTS

Title	No. required per shift		Duties
	1 unit	2 units	
Boiler engineer [1]	1	1	Responsible for the proper operation and maintenance of all boiler-room apparatus and coal and ash-handling systems.
Boiler operator	1	1	In general charge of boiler operation during his shift. Reports to boiler engineer.
Stoker operator	3	6	Each operates two boilers, including stokers, clinker grinders, dampers, etc.
Soot-blower operator	1	1	Operates soot blowers on each boiler once during the shift.
Larry operator	1	1	Weighs and distributes coal to stoker hoppers, keeping all active boilers supplied. In order to provide a consistent record for daily heat balance, hoppers on all active boilers are filled full between 11:30 and midnight.

[1] Day shift only.

200,000 gal. and serves to take up fluctuations in the demand for feedwater and to provide a reserve supply for emergencies.

51 The evaporators have in general given satisfactory service, although some trouble was experienced due in part to improper design of some features and in part to lack of experience with the particular water being evaporated. They are so connected that either river or deep-well water may be used. The river water, although very dirty, was found to contain less scale-forming salts than that from the wells, and is therefore generally used. The effects are reversed every four hours, thus minimizing scale formation. The thermal efficiency of the evaporators is practically 100 per cent, and with the modifications now being made it is thought they will give no trouble.

52 Water qualities throughout the cycle are closely watched. Table 7 shows water analyses at various points in the system. A multiple-point indicating conductivity meter with cells located at various points is used to observe the water purity continuously. This instrument immediately indicates leakage of circulating water into the condenser and permits a close check in the quality of the distillate from the evaporators. No condenser leakage has as yet

TABLE 10 COAL- AND ASH-HANDLING FORCE, ONE TEN-HOUR SHIFT ONLY

Title	No. required		Duties
	1 unit	2 units	
Coal Handling			
Coal foreman	1	1	In charge of coal handling to bunker.
Coal-conveyor operators	2	2	Operate crushers and conveyor system
Coal laborers	2	2	Dump coal cars.
Ash Handling			
Crane operator	1	1	Operates locomotive crane in assisting to distribute ash; part time only.
Locomotive operator	1	1	Operates dinky locomotive in shunting coal cars; part time only.
Laborers	2	3	Load ashes to cars.
Miscellaneous			
Coal and ash sampler	1	1	Takes and mixes samples, etc.
Boiler Maintenance			
Boiler repair man	2	3	Repair leaks, clean inside of tubes, etc.
Boiler repair man helper	1	2	
Stoker repair man	2	2	Repair and overhaul stokers.
Stoker repair man helper	2	2	
Bricklayer	1	1	Repair brickwork
Bricklayer helper	1	2	
Furnace cleaners	2	2	Clean slag from tubes, etc.
Soot-Blower Maintenance			
Pipe-fitter	1	1	Repair soot blowers.
Pipe-fitter helper	1	1	

been detected, but much useful information on the performance of the evaporators has been obtained.

53 It has been found that dissolved oxygen in the feedwater tends to pit the boiler drums. It has also been determined that this pitting takes place during banked periods, there being practically none when the boiler is active. In order to decrease the quantity of dissolved oxygen to the lowest possible point, all parts of the feed cycle are effectually sealed against air. The feed temperature is maintained at 205 deg. or higher to drive out the entrained air.

Further experiments are being carried on to determine whether a lower feed temperature with a higher vacuum in the barometric condenser will improve this condition.

54 Fig. 17 shows the relation between temperature and oxygen content as observed at Colfax. Table 8 gives the results of observations of dissolved oxygen in the boiler feedwater taken from the condensate head tank and the boiler-feed tank.

55 Tables 9 and 10 show the personnel of the boiler-room organization with one unit as at present, and as it will be when a second unit is installed.

56 Table 11 is a summary of boiler-room operations for the month of December, and Table 12 is a recapitulation for the six months August to January, inclusive.

TABLE 11 SUMMARY OF BOILER DATA — DECEMBER, 1921 — COLFAX POWER STATION

Boiler No.	Lb. Water Evaporated	Lb. Coal Consumed	Hours Active	Efficiency, per cent	Rating, per cent	Exit Gas Temp., deg. fahr.	CO_2 per cent	Combust. in Refuse, per cent
1	60,020,000	6,638,060	559	76.6	172	489	9.9	26.67
2	55,291,000	6,089,230	480	76.9	185	482	12.0	26.89
3	75,435,000	8,319,030	696	76.7	174	460	10.3	28.60
4	75,542,000	8,324,290	691	76.8	175	481	11.9	25.71
5	32,242,000	3,519,710	285	77.5	183	—	11.5	29.55
6	73,474,000	8,138,360	688	76.5	171	479	11.7	26.88
8	70,546,000	7,761,900	640	77.0	171	481	11.0	27.18
Total	442,550,000	48,790,580	4038	76.8	176	478	11.2	26.99

Average superheat, 132 deg. fahr.
Average feed temperature, 193 deg. fahr.
Average B.t.u. in coal, 13,283 per lb.

57 The heat balance for the station is worked out daily. Table 13 shows the averages of the heat balance figures for the months of October to January, inclusive.

CONCLUSION

58 In considering the results given in the paper the following points should be borne in mind:

a The operating results given are from the records of the operating department of the Duquesne Light Company

b The plant is not equipped with heat-reclamation equipment such as economizers or air preheaters

TABLE 12 SIX MONTHS' SUMMARY OF BOILER DATA — COLFAX POWER STATION
AUGUST, 1921, TO JANUARY, 1922, INCLUSIVE

	August	September	October	November	December	January	Average
Total water evaporated, lb.	272,779,000	265,373,000	292,355,000	248,858,000	442,550,000	313,703,500	305,936,417
Total coal consumed, lb.	30,085,600	28,023,100	31,690,300	27,109,300	48,790,600	35,911,530	33,601,738
Total active service, hours	2,782	2,493	2,474	2,361	4,038	2,894	2,840
Average feed temp., deg. fahr.	208	210	203	204	193	204	204
Average superheat, deg. fahr.	124	120	127	127	132	136	128
Average B.t.u. in coal per lb.	13,093	13,180	13,159	13,198	13,283	13,177	13,182
Average efficiency, per cent	76.4	78.9	78.0	77.2	76.8	77.6	77.5
Average rating, per cent	154	167	187	167	176	168	170
Average exit gas temp., deg. fahr	466	460	465	467	478	471	468
Average CO_2, per cent	9.2	9.3	9.6	10.3	11.2	11.1	10.1

c The efficiencies given are not test results but have been made with the regular operating force and can be maintained year after year. The results are good but do not represent the best performance possible with the equipment. Some further improvement is expected as the operating force becomes more familiar with the equipment

TABLE 13 TYPICAL HEAT BALANCE DATA — COLFAX STATION

	October		November		December		January	
Coal as fired, B.t.u. per lb.	13,159		13,198		13,283		13,177	
Ash, as fired, per cent	9.28		9.37		9.14		9.05	
Moisture, as fired, per cent	4.45		4.12		3.91		4.49	
Inlet air temp. deg. fahr.	70		70		70		70	
Exit gas temp. deg. fahr.	465		467		478		471	
Carbon as fired, per cent	72.29		72.49		72.86		72.45	
Hydrogen as fired, per cent	4.92		4.93		4.96		4.93	
CO_2, per cent	9.5		10.3		11.2		11.1	
Combustible in refuse, per cent	26.36		29.10		26.99		26.76	
	B.t.u.	per cent	B.t.u.	per cent	B.t.u.	per cent	B.t.u.	per cent
Heat absorbed by boilers	10,264	78.00	10,189	77.20	10,202	76.8	10,225	77.60
Moisture loss	55	0.42	51	0.38	48	0.36	56	0.42
Hydrogen loss	545	4.14	547	4.14	522	4.16	548	4.15
Stack loss	1,713	13.02	1,585	12.01	1,522	11.46	1,502	11.40
Ashpit loss	482	3.66	558	4.23	490	3.69	479	3.64
Heat accounted for	13,059	99.24	12,930	97.96	12,814	96.47	12,810	97.21
Heat unaccounted for	100	0.76	268	2.04	469	3.53	367	2.79
Total	13,159	100.00	13,198	100.00	13,283	100.00	13,177	100.00

d The amount of combustible in the ash is high, but as explained above, the necessity for early quenching of the ash to prevent clinker makes improvement in this respect doubtful. In the light of overall efficiency the slight excess in combustible is a lesser loss than the formation of hard clinker masses would occasion.

DISCUSSION

T. E. Keating. The paper is a valuable addition to central-station engineering, showing what excellent results can be obtained with modern high boilers and under-feed stokers without the aid of heat reclaiming apparatus such as economizers and air preheaters. This tabulation probably represents the highest operating performance published and is particularly remarkable inasmuch as the valves given are an actual record of the monthly performance from the station data. These efficiencies should not be compared with a boiler test in the common interpretation of the term, which covers merely the performance of one isolated boiler operating for some hours at a predetermined rating with no allowance for such losses as banking, emptying boilers and other losses incidental to boiler-room operation which tend to lower the overall operating efficiency.

The gradual improvement of efficiency from the time of installation is not due to any vital changes in design of either the stokers or the boilers, but is the result of careful study of the combustion of the fuel employed with reference to the particular boiler-room equipment and operating demands upon it, followed by the development of an operating organization trained to carry out the routine determined by this study.

The combustible in the refuse looks high, but it is difficult to get a representative sample of refuse and, therefore, somewhat difficult to obtain an absolute value for this factor. In any practical stoker the combustion is never complete and the degree to which complete combustion is approximated is a compromise of thermal and mechanical conditions which give the most economic operating condition. It is possible to obtain more nearly complete combustion by installing more grate area, but the excess air required beyond a certain stage more than offsets the gain in efficiency due to the reduced combustible in the refuse. The nature of the coal burned is a vital factor in obtaining proper combustion and the injection of water has proven the most satisfactory method of eliminating clinker formation with Pittsburgh coal. Clinker trouble is almost completely eliminated and it has not been necessary to grind out the pits since last August.

Undoubtedly the CO_2 is somewhat higher than indicated by the station instruments, as it is difficult to get a true average value of this item over so large a fuel bed. Isolated readings with test instruments have indicated values above 15 per cent, and the efficiencies that do exist in this boiler room are evidence that the CO_2 is undoubtedly higher than the average value indicated by the station instruments.

There has been a small amount of slag as a result of soot-blower elements burning off. This has been corrected as the author states, but it still may be desirable to install more soot-blowers in order to be certain of entire elimination of trouble from this source. There is a hard scale that forms on the outside of the tubes which is not exactly a slag, as it is not sufficient to shut off the gas passage although it does act as an insulator. This apparently first forms a powder and then hardens. It is soluble in water and when the boilers are off for cleaning, water is applied to the outside of the tubes and the scale flakes off.

While the average rating on these boilers very nearly corresponds to the point of highest efficiency on the boiler, nevertheless, any individual boiler operates only a few hours per day at this rating. For the greater part of the day, it is operating either above or below the point of best efficiency, which, of course, reduces the daily average efficiency to a point below that corresponding to a steady load at the average boiler rating.

Fig. 13 shows that recently No. 5 boiler is materially better than the others. This is due to the fact that various minor adjustments, such as changes in damper control, relocation of soot-blowers, and modification of baffles have been combined in this boiler while it was out of service. Similar modifications are to be made in the other boilers when they are taken off for overhauling and it is expected they will show corresponding improvement.

The time required for a boiler to reach the point of maximum efficiency has been reduced from over seven days to under two days, due principally to better air control. The boilers are started with a minimum air supply and the heat is at first more or less bottled up so that it is not necessary to spend so much time heating up the settings. This reduction of time in obtaining maximum efficiency will naturally improve the net monthly efficiency and this latter value will also be further improved by some refinements in damper control that are now being experi-

mented with. Originally maximum efficiency could be maintained for about three weeks before it started to drop. Now that it is possible to get up to the point of maximum efficiency in two days and to maintain this value for over two months, and because of various minor operating improvements under way, there is no reason why the present high efficiency cannot very easily be maintained and even improved.

B. N. Broido. This paper is most timely. A number of large power stations are now under consideration or construction; boilers between 1500 and 2000 hp. seem to be the most preferred, and the performance of the boiler at the Colfax station is of considerable interest, and the members of the Society should be indebted to the author for making this information available.

The design of boilers has been changed very little in the last years, and it seems that all efforts are concentrated in improving accessories, such as stokers, superheaters, economizers, etc. The relation between the first two is more significant than usually recognized. As a matter of fact, the capacity and performance of a superheater is largely dependent upon the conditions in the furnace, and also upon the relative location of the two.

As the Colfax station is going to have superheaters in practically identical boilers, but in two different locations with respect to the furnace, it will be interesting to observe the performance of the superheaters. In general, the performance of a superheater is influenced by the velocity and the temperature of the gases. In a superheater located as at present at the Colfax station, the velocity of the gases is comparatively high, while the temperature is rather low, and most of the heat is absorbed by convection. In this case with increased rating, the heat absorbed increases in greater proportion than the rating itself, so that the superheat is higher at the higher rating, than at the lower.

With the superheater located near to the furnace where the temperature is very high, the heat is absorbed by radiation. In spite of the fact that radiation increases with the difference of the fourth power of the absolute temperature, the heat absorbed by radiation does not increase or decrease in proportion to the rating. A superheater can, therefore, be so located in a *low velocity* and *high temperature* zone, that the heat absorbed by radiation and convection is so balanced that the superheat will remain fairly

constant over a large range of rating. Such superheaters have been built, and are now in operation, as for instance, the superheaters in the large Hell Gate station, where a fairly constant temperature is obtained for all ratings between 100 and 300 per cent. This is possible particularly with large boilers provided with large combustion spaces.

On Fig. 8, which illustrates the boiler, it is stated that the heating surface of the superheater is 6700 sq. ft. It is mentioned that some trouble is experienced with the superheat being less than specified. This is somewhat misleading. One may derive the impression that 6700 sq ft. of heating surface is not sufficient for 180 deg. superheat, while as a matter of fact, there are a number of superheaters in the same location, and in the same proportion to the boiler, or even smaller, giving 300 deg., and more, superheat. The superheat in the Colfax station is probably of the so-called "extended-heating-surface" type, the elements of which consist of steel tubes covered by cast-iron rings with ribs, and the heating surface is, no doubt, that of the outside surface of the cast iron. Naturally, this heating surface cannot be compared with the heating surface of a boiler tube, or of a superheater with bare tube elements.

The resistance to the flow of heat from the gases to the steam in a bare tube superheater consists of three components — the resistance between the gases on the metal wall, the resistance through the wall, and thirdly, from the wall to the steam. In the extended-surface type of superheater, two more components are added — the resistance from one metal to the other, and the resistance of the cast iron. The resistance to the heat flow from one metal to the other depends upon the condition of the surfaces of contact; also upon the pressure between them. The writer has made experiments with the flow of heat between two tubes with different contact surfaces and different pressures, and the results will be published later. As the contact between a steel tube and a cast-iron cover subjected to high temperature will not remain always metallically clean, it offers considerable resistance to the heat flow. The temperature of the outside cast iron is then considerably higher than the steel tube, and the heat absorbed is, therefore, less.

It would be interesting to know what is the steam-touched heating surface of the superheaters.

Of considerable interest is the experience, mentioned by the author, with air and oxygen contained in the feedwater. This information is most timely now, when the steel economizer seems to be coming into its own. In his experience with steel economizers, the writer has found that when the feedwater is heated in an open feedwater heater, there is a considerable difference in the oxygen content of the water, for the same pressure and temperature, whether the feedwater heater is vented often, or not. If it is entirely closed, considerably more oxygen can be contained in the water than if often vented. The feedwater temperature of 205 deg. seems to indicate that the vacuum in the barometric condenser is very low. The Society would certainly be grateful to the author for information regarding the arrangement and system of heating and degassing the water by the barometric condenser.

Nevin E. Funk. The author has presented a very comprehensive paper on the boiler-plant-design characteristics of the Colfax plant. As is ever the case, no two men look at anything from exactly the same angle, and many factors entering into plant construction are merely matters of individual preference. There are, however, some basic principles that should not be violated.

One of these, and not the most unimportant, is that the simplest control of operations should be obtained that is consistent with the best results. With this in mind the writer cannot quite see the necessity of two individual motor drives on the clinker crushers. The result of this installation is that the operator must vary three motors instead of one. Of course, when the clinker grinders are operated intermittently this may not be so great an objection.

From the operation of two stations with clinker grinders that come under his control the writer is convinced that the best results are obtained by continuous operation of the rolls, the speed of the rolls varying with the stoker speed but with adjustable ratios at will. With proper ratio settings the ash is removed at the same rate that it is formed and furnace conditions kept very uniform. It must be borne in mind that this type of crusher drive must come from a separate reduction gear driven by the motor counter shaft and not from an extra crankshaft on one of the ram-box gears, the latter type drive having some serious defects.

If it has been found desirable to operate the Colfax roll inter-

mittently the writer would judge that the rolls cannot be operated slowly enough and therefore if the gear ratio were increased by 4 to 1 advantages in continuous operation would be found. The writer unfortunately had this very experience with his first clinker crusher installation.

Referring to Par. 23, lest the idea be got that it is impossible to operate clinker crushers without drowning them, the writer desires to take another leaf from his own experience. The first crusher-pit design which the company with which he is connected installed caused endless trouble until a large amount of water was sprayed into the pits. By careful tests and observations the causes of this trouble were determined and in later designs absolutely eliminated. In a plant of this company there are now eight 1500-hp. boilers than can be operated at 1100 lb. per ram-hour; and since the installation was made in 1920 not one drop of water has been put in the clinker-crusher pit.

Two reasons for the necessity of water in the Colfax crusher pits are first, they are too shallow; second, the V-shaped sides act as the buttress for an arch and the clinker will flux across this and hang up unless kept cool by spraying. The difference between using water and not using it is a reduction from 25 per cent combustible in the ash to 10 per cent.

The writer would like to ask why no stoker-speed indicator (rate of coal feed) was installed. We find this very valuable indeed.

Referring to Figs. 13 and 14, some unusually high operating efficiencies are plotted. The length of gas passage, some 60 feet, would account for some of this, but hardly for the efficiencies shown by No. 8 boiler in September and No. 5 boiler in January and February. The writer understands that these are the figures as taken from the meters and coal scales, but would like to know if the author has any data as to the possibility of error in these particular water meters on these dates, and if not, if there were any particular conditions of operation on the two boilers that resulted in these high efficiencies. If not, and if it can be done for a time on some boilers one would think it could be done all the time on all boilers with careful operation.

In Par. 39, attention is called to the fact that the superheat was low by about 35 deg. fahr. This seems to have been quite prevalent in the latest installations where very large furnace

volumes were installed. Apparently with the lower set boiler, secondary combustion played a considerable part in producing superheat as well as the larger gas volumes that were used per pound of coal. With the advent of more efficiently designed furnaces which are also more efficiently operated the heat available at the superheater as well as the heat transfer rate are both reduced and more surface or a re-arrangement of gas passages is necessary to obtain the results expected.

The curve for dissolved oxygen shown in Fig. 17 is about what is to be expected at atmospheric pressure. The writer would like to know if any experiments have been made on operating the barometric heater with a vacuum so that lower feedwater temperature may be obtained and the oxygen reduced to approximate zero.

Finally, the heat balance shows remarkably low unaccounted-for losses, which speaks very well for the design of the walls as insulating media.

With reference to the question of oxygen absorption, it is probable that many will be surprised to find the actual oxygen content less than the theoretical. The curve shown in Fig. 17 is for the theoretical amount of oxygen to saturate water, that is, the maximum amount of air that can be contained in water at a given temperature.

To obtain this condition requires time and complete exposure of the particles of water to the atmosphere. If the water is moving in a rapid cycle it cannot possibly take up the amount of oxygen necessary to saturate the water. In addition, if there is any possibility of the water containing sewage, this will absorb part of the oxygen so that it will not appear in the oxygen test. The use of chemicals for boiler-feed treatments, if of the proper constituents, will also reduce the measurable oxygen content. These things all have a tendency to make the oxygen appear of a lower value than the theoretical saturated value.

D. Robert Yarnall. In Fig. 16 is shown a comparison of V-notch and venturi-meter records in which the venturi readings have been used for comparison basis, although the average variation of plus 1.3 per cent shows very close coincidence of these two methods of measurement. Why were not the V-notch readings used as the basis of measurement?

In the writer's experience in the measurement of hot boiler

feedwater, which covers a period of the past ten years, he has felt that the V-notch weir method of measurement is extremely dependable because it is accurate at low rates of flow as well as high, and moreover, because it compensates automatically for variations in temperature, because of the hydrometer action of the float which operates the recording mechanism.

The author is to be congratulated upon having presented a paper which really gives the performance of one of our great stations. In the past, such performance figures have not been given sufficient prominence, and it is hoped that this paper will be the forerunner of similar presentations of performance of other notable stations built in the last few years.

A. T. Hutchins. What is the maximum and minimum rating at which the boilers are operated for each day? Is the CO_2 the weighted average, which would possibly explain Mr. Keating's point?

E. A. Trinks. Is it possible with one control system to handle the air, as well as the coal supply, as it is fed into the stokers; in other words, does the coal supply to the stokers vary directly in proportion to the air used?

H. H. Bailey. Is there a possibility of overheating the superheater in its present location?

W. A. Carter. In measuring the temperature of the flue gases leaving the boiler were the terminals shielded against radiation with the cooler surface? It is possible in some places to have errors of 50 deg., the indicated temperature being that much lower than the actual temperature of the gas.

E. K. Davis. Was trouble experienced in the fusing of the nozzles of soot blowers?

Is the baffle the poured or is it the ordinary B.&W. type of baffle? If the poured type of baffle was used, was any trouble experienced in flaking?

Paul Yopp. Can the author give us any information as to the barometric degassifier?

The Author. In reference to the comments of Mr. Funk, about the double drives on clinker grinders; these grinders are

about 30 ft. wide, and at the time of designing them, it was thought best to drive from both sides. The operation resulting from this design has justified the designers' ideas on that subject.

As to the speed of the rolls, we never have been able to run the rolls slowly enough. Running the rolls even as slowly as four revolutions an hour is too fast. For continuous operation the rate of speed of the clinker grinder depends entirely on the ash bed over it. If it is ground at too great a speed, the clinker grinder is burned out. We expect to make an improvement in the use of water in the pits. It has been found absolutely necessary to maintain flowing water in the pits to avoid boiler "outages" and other troubles due to clinkers.

To touch again on the comparative efficiency of boilers Nos. 5 and 8, we have had an actual difference in this plant of over 10 per cent between adjacent boilers. When ferreted out, we find a certain type of ram, tuyers or ram throat which is less efficient than another. The boilers are not identical, neither are the efficiencies identical. In any other boiler plant the size of this, two or three per cent difference in efficiency can be found in the boilers if operated at the ratings given. The man who is running a particular boiler has something to do with this difference. We have never found any of the meters on the boilers with an inaccuracy of over one per cent. The average difference in accurate value of the meters by actual test has been about 0.07 and never over 1 per cent.

The superheat is low in the original boilers due to heat absorption through the machine prior to the superheater.

As to the lowering of oxygen in the barometric heater, we are trying some experiments in putting a slight vacuum on the barometric head, to see what can be done in the way of dilution which will remove the air, and also in steam blanketing the feed tank where most of the absorption takes place.

At the time the plant was built coal was selling at $2.35 at Pittsburgh. We realized that the price of coal was going up, so that $3.50 was the rate assumed for a period of ten years. At that rate, and with the construction costs prevailing at that time, the economizers would begin to pay for themselves when coal reached $7.00 a ton

There were two questions raised regarding oxygen elimination. The oxygen elimination in the heat-balance system involves two

factors, first, prevention of absorption, and second, driving out the oxygen by raising the temperature of the feedwater. It is perfectly natural that the oxygen content should be lower than the theoretical value in a cycle of operation such as this, due to the time element which is necessary for absorption.

In answer to Mr. Hutchins' comments on the maximum boiler rating, these boilers operate normally from 130 to 210 per cent of rating during the day of 24 hours. We have run them on test as high as 313 per cent for four hours, but the average would be around 175 per cent.

For February, March and April the total output of the plant was 97,575,100 kw-hr., the load factor 77.5 per cent, and the B.t.u. rate per kw-hr. output for these months was 18,713.

Replying to Mr. Yarnall's remarks, reference to the paper will show that we have given a comparison of results between V-notch and venturi meters. The venturi meter readings were used merely because of the author's preference for this type of recorder. It will be noted from the curve in the paper, however, that there was very close agreement between the two methods of measurement.

In answer to Mr. Trinks' comment as to air and coal control, some persons think that coal and air can be controlled automatically. It cannot be done. The air is controlled automatically, and the coal by hand.

In answer to Mr. Bailey's comment regarding the overheating of the superheaters, in the new location it would be impossible to overheat the superheater unless the flow of steam was interrupted.

As to Mr. Carter's question on the measurement of flue gases, we use mercurial pyrometers for these readings. We have tried others, but not with satisfactory results.

In answer to Mr. Davis' questions, no difficulty has been encountered with soot-blower nozzles. The baffles are the standard B.&W. baffles, brick-backed, with cast-iron supports. We are going to try plain cast-iron baffles on some additional boilers. We have not used any poured baffles in this station.

No. 1847

CENTRIFUGAL CASTING

By Leon Cammen,[1] New York, N. Y.
Non-Member

The paper describes the process of centrifugal casting of hollow metal objects. The author first takes up the field of centrifugal casting and the history of the development of the art, which he follows by a discussion of the mechanics of the problem and a description of the operation of the casting machine. He then takes up the thermal conditions in the mold and the field of application of hot-mold centrifugal casting, and closes with a discussion of the manufacture of plates by the centrifugal casting process.

In appendices to the paper are given a bibliography of the subject, a list of patents, some of the mathematics covering the cases of casting about a vertical and about a horizontal axis, and the forces acting on the liquid metal in the direction of the axis of the mold, a discussion of temperature control in the casting and in the mold, and an analysis of the stresses in the mold.

1 The art of centrifugal casting of hollow metal objects is quite old and has been practiced on a commercial scale since the beginning of the last century. The earliest English patent (Eckert) dates as far back as 1809, and the earliest American patent to Lovegrove was issued in 1848. At about the same time Andrew Shanks, in London, England, began to make cast-iron pipe 12 ft. long and 3 in. in diameter by pouring molten metal into a spinning wrought-iron mold. His process was described in America in the *Scientific American* of December 1, 1849, and it is of interest to note that in its basic features of design the Shanks machine does not differ in any way from the great majority of machines working with a cold mold at the present day.

2 Attempts were made at an early date to apply centrifugal casting to ingot making with the view to improving the quality of the metal and, for example, M. Tresca, in a paper before the Institution of Mechanical Engineers,[2] states that the most remarkable

[1] Associate Editor, *Mechanical Engineering*.

[2] *Proceedings*, 1867; pp. 149–150.

Presented at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

instance that he had met with of freedom from air bubbles was in the case of a bessemer rail manufactured at the Imphy Iron Works near Nevers in the Department of Nièvre; and he had ascertained that the method adopted at these works was to pour the molten metal into a revolving vessel driven at a considerable speed so as to clear the steel of air by rapid rotation. This method proved very effective in practice in freeing the steel from the minute air bubbles which it contained on leaving the converting vessel or the melting pot.

3 A survey of the literature (see Appendix No. 1) and of the numerous patents issued on centrifugal casting in this country (Appendix No. 2) and in Europe would indicate that practically all the essential features were covered by patents issued prior to the beginning of the twentieth century, with the exception, however, of those dealing with the temperature control of the metal, which is a new feature. It would appear, therefore, that with the exception of minor details and one or two features in special casting processes there is no reason why any good engineer could not design and operate a successful casting machine with a non-heated or non-cooled mold without running into legal complications. At the same time, it might be well to realize that the very fact that the basic features of the general process are already free makes those who have developed some of the few minor features particularly anxious to protect their patent rights in them. Because of this a general familiarity with the patent situation is desirable to avoid involuntary infringement by using some minor detail such as, for instance, a particular type of pouring spout that has been covered by a patent. It is to assist in this that Appendix No. 2 has been prepared, giving as far as the author is aware, a complete list of all patents on centrifugal casting issued since 1848 by the U. S. Patent Office and prepared after a careful investigation of the entire field. (See also Appendix No. 1, Bibliography).

MECHANICS OF CENTRIFUGAL CASTING

4 In the first place, a clear distinction should be made between casting about the horizontal and about the vertical axis and also the intermediate case of an inclined axis. In casting about a horizontal axis the metal is distributed symmetrically about the axis of rotation, and if the axis of rotation coincides with the axis of the spinning unit, a tube of uniform thickness is produced for the entire length of the mold. If a vertical axis is used it is obvious

that the external wall of the casting follows the shape of the mold while the interior of the casting forms a paraboloid of revolution, the elements of which depend on various factors given in Appendix No. 3. In casting with the inclined axis the same general principles apply, with the further complication that the elements of the inside paraboloid of revolution are also affected by the angle of inclination of the axis to the true horizontal.

5 The usual way of casting is by introducing molten metal into the spinning mold. Where the chilling of the metal is extremely rapid, as, for example, in casting cast-iron pipe against a water-cooled chilled mold, it is imperative to use a movable spout, the latter sliding at a certain predetermined rate so that by the time the nozzle discharging the metal comes out of the mold the entire pipe is completed. This is the process employed by DeLavaud. Other manufacturers working with the cold mold have attempted to secure the same results by what is known as a trough spout, which is really a trough into which the entire metal of the casting is poured at a rapid rate. The trough is then tipped and discharges the metal suddenly, the metal falling on to the walls of the mold at such a rapid rate that a pipe forms before the metal has had time to chill. Hitherto this process has met with only indifferent success.

6 A better result has been achieved on comparatively short castings, however, by the use of a ledge spout, the spout ending in a flat ledge along which the metal can flow the entire length of the casting. Finally, where a hot mold is used an ordinary short spout is employed, the metal distributing itself throughout the mold longitudinally under one of the components of centrifugal pressure or pressure produced by centrifugal action (Appendix No. 3). This arrangement cannot be used with the cold mold for reasons explained in the Appendix.

7 In centrifugal casting the distinction should be clearly made between jobbing work and mass production. For the former, clay molds in iron cases have been used practically exclusively, as it would not pay to make a permanent mold for only a few castings. When it comes, however, to production on a tonnage basis, such as tubing or pipe, a permanent mold is practically the only feasible way of doing it. Crucible graphite or zirconia tubes have been successfully used for castings not in excess of 3 ft. in length, but when it comes to longer ones, metal molds are to be preferred. Fairly successful results have been obtained with chrome-nickel

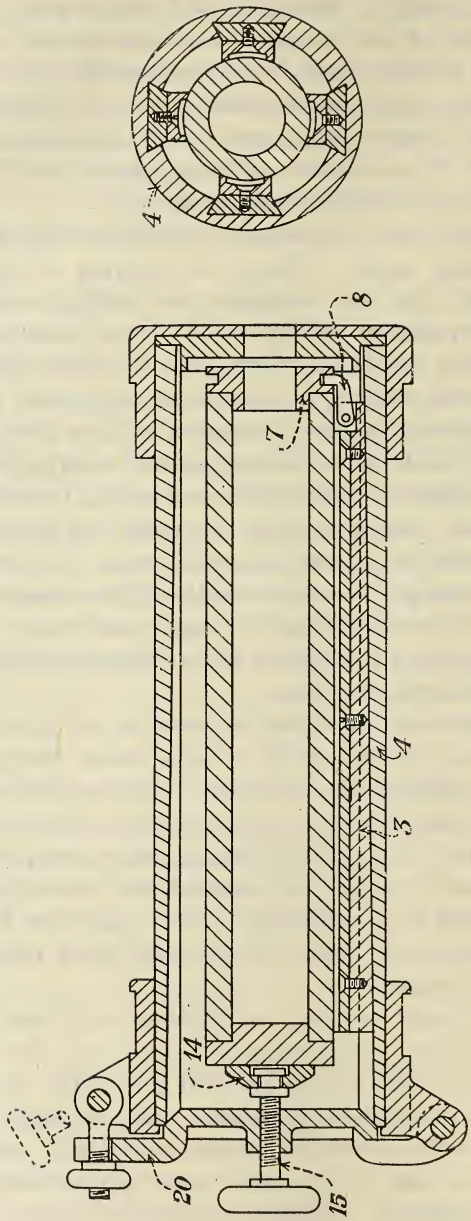


Fig. 1 Spinning Machine for Use With Removable Molds, Heavy Type

alloys, though it is not expected that this will be the material finally adopted, as enough work has been done already on the production of cast tungsten to make it certain that a mold of this still rather mysterious material will be available within a comparatively short time.

8 There are several ways of supporting the mold in the spinning bench, depending on whether the mold is a permanent fixture or removable and also depending on the size of the mold. Where the mold is a permanent fixture as it is today wherever a cold mold is used, any kind of bearings, providing they are substantial enough, may be used. With removable molds the conditions are somewhat different, especially where the mold is heated to a high temperature. (It will be shown later that the temperature of the mold at the time of pouring the metal may be as high as 2000 deg. fahr. — see also Appendix No. 4.) Where extremely hot molds

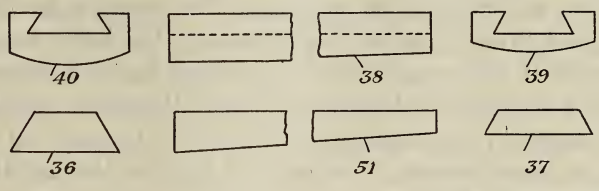


Fig. 2 Gibs and Beds of the Spinning Machine Shown in Fig. 1

have to be used, the arrangement must be such that the stresses on the mold can be reduced to a minimum, and that the mold can be rapidly and easily inserted into the machine and extracted therefrom. The matter of stresses is of very great importance in view of the fact that, especially in large machines, the bursting stresses due to the action of centrifugal force are of such magnitude as to require the most serious consideration. (Appendix No. 5).

9 The spinning bench for such removable molds works on a comparatively simple principle, the mold being held in a barrel rotating on rollers. The barrel construction shown in Fig. 1 is particularly suitable for use with hot molds. Here the barrel consists of a steel shell, 4, which may be an extra heavy steel pipe, but in sizes above 12 in. has to be made as a special casting. At four to six places symmetrically spaced along the inner circumference of the shell, steel strips tapered at 39 and 40, Fig. 2, and with face machined to the same radius as that of

the internal surface of the shell, are screwed on or riveted on. The strips are provided with truncated slots as shown in the figure, and in these slots move the keys or gibs 51, the cross-section of which is shown by 37 and 36, which indicates that the face 36–37 is of the same width from one end of the gib to the other, but the height of the trunk varies.

10 It is obvious that as long as the four gibs move through the same distance longitudinally in their tapered beds, the faces of the gibs will remain parallel to each other but the enclosed cylinder tangent to those faces will vary in diameter with the position of the gibs.

11 The operation of the machine is therefore as follows: The head, 20, is opened and the gibs pulled out a little way until the tangent cylinder is, say, a quarter of an inch larger than the external size of the mold. The mold is then inserted, preferably without touching the gibs, until it comes to bear against the abutment ring, 7, to which all the gibs are attached, (for example by hooks, 8). As the mold presses against the abutment the latter recedes and carries with it the gibs, which by moving in their tapered beds gradually contract until they come to grip the mold, and they may be made to grip it as hard as desired simply by exerting sufficient pressure on the abutment, 7, which may be done through the block, 14, by the screw, 15.

12 The important advantage of this construction lies in the fact that, first, the mold automatically finds its own center, and, second, that it is supported all along its length, a matter of great importance when we come to deal with the bursting stresses in the mold. (Compare Appendix No. 5.) Whether a solid or a split mold should be used is a matter determined primarily by the shrinkage condition of the metal of the castings and the metal of the molds. Where the shrinkage conditions are such that the casting comes out of the mold easily, that is, where it contracts quicker than the mold, a solid mold may be used, but where the shrinkage conditions do not guarantee easy extraction of the casting, or where the shrinkage conditions are non-uniform, the use of a split mold has been found advisable. Such molds are usually split longitudinally and the question of the best method for holding the two parts together is far less simple than it appears at first sight. There are numerous patents showing the two split parts of the mold held together by bolts. In actual construction, however, the use of bolts for this purpose is entirely unsuitable, as may be shown by simple calculations.

13 To meet this condition the mold shown in Fig. 3 has been developed. This mold has side lugs machined in such a manner that when the two parts are put together a tapered structure, such as 6, is obtained, and dovetail pieces, 5, are provided to engage with the taper 6. It is obvious that when the dovetail piece, 5, is driven hard over the lug taper, 6, it will hold it tight. Furthermore, both the lugs and the dovetails can be made within reason practically as heavy as desired, so that a good factor of safety can be provided to take care of bursting stresses in the mold. To close or open the mold, it is only necessary to apply pressure to the dovetail piece or give a light blow, and by means of a simple jig the blow or pressure can be applied to all the dovetail pieces on the mold at once.

14 Such a construction diagrammatically is shown in Fig. 3. It is obvious, however, that the presence of the lugs makes the machining of the mold to a true cylinder expensive, if not impossible. Because of this, instead of machining the mold to a cylinder, a number, such as four, of flat faces are planed on the mold, the location of the faces being such that they will be distributed symmetrically around the longitudinal axis of the mold and that they will be tangent to a circle described with the point on the axis of the mold as a center. Such an arrangement is comparatively easy to carry out and it gives an excellent support for the mold, besides providing a structure a good dynamic-balance properties.

15 As regards the foundations and bearings for centrifugal-casting machines no special design need be described here, but it should be most clearly and strongly emphasized that both the bearings and the foundations should be of very generous proportions and most substantial design. The machine in centrifugal casting is subject to quite violent strains and unless it is properly built it is apt to get into vibrations which sooner or later may have a dangerous effect. The writer knows of a case in an American plant where a centrifugal-casting machine located in a ditch and built with insufficient ruggedness actually jumped out of the ditch during operation, endangering the lives of those around it. With steel and concrete amply available there is no excuse for such practices.

16 Up to about two years ago very little work, as far as the writer knows, had been done successfully on the problem of casting centrifugally comparatively thin sections, say, under $\frac{3}{8}$ in., in such metals as steels or monel and in lengths beyond 3 ft. On the other

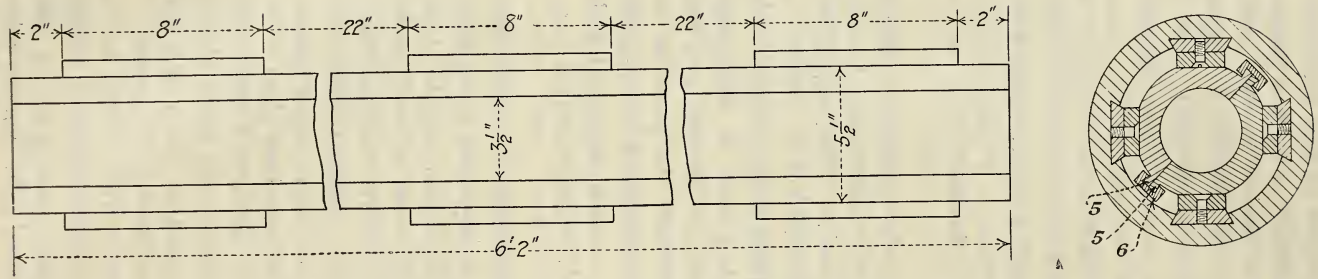


Fig. 3 Variation of Fig. 1, With Split Mold Held Together by Tapers and Dovetails

hand, large castings in brass, bronze, cast iron and even steel have been made with considerable success. The reason for this lies in the failure to understand the thermal conditions in the mold and the process underlying the freezing of the metal.

17 There is a basic difference between castings in a stationary mold and in a spinning mold. In making castings in the former, whether it be a simple ingot or a complicated casting from a pattern, the foundryman has means to provide for the escape of gas from the metal and for taking care of the cavities produced by contraction in cooling. In a casting from a pattern, sink heads are provided, while in an ingot casting the top of the ingot acts as a sink head, in addition to which special methods such as "dozzling" may be used to keep the top of the ingot hot as long as possible, and increase the efficiency of the ingot top in performing the functions of the sink head in molds. In both cases the foundryman is prepared to discard as defective a certain part of the casting (the ingot head, risers, sink heads) and his skill lies partly in reducing the quantity of discard as far as possible and especially in directing the process so as to restrict the presence of defective metal to the parts which he is prepared to discard.

18 In centrifugal casting, there is no riser or

anything corresponding to the ingot top, and the cooling proceeds essentially at a uniform rate all along the length of the tubular shape. It is obvious, especially in tubes of small diameter and considerable length, such as 6 in. outside diameter and 16 ft. long, that the loss of heat inward must be extremely small, because the entire inner air cylinder is surrounded by metal at approximately the same temperature. Heat is therefore lost mainly outward to the mold and through it to the air. As a result of this, the part of the casting in immediate contact with the mold chills first. The problems with which the " centrifugal " foundryman has to deal are essentially the same as those which confront the man pouring into stationary molds, and these are, first, to get rid of the occluded gases, and, second, to take care of contraction cavities. As regards occluded gases and slags, it would appear at first sight, especially to those familiar with such centrifugal processes as cream separation, that they would be eliminated automatically; pure metal, being heavier than either slag or metal containing gases, should be thrown to the outside, against the wall of the mold, while all impurities should go to the inner wall. This does actually happen, provided one condition is satisfied, and that is that sufficient time is available; and it should be remembered in this connection that this separation takes place at a fairly slow rate because of the great viscosity of molten steel.

19 The second problem is that of cavities formed in the casting as a result of contraction in cooling. As stated above, the part of the casting in immediate contact with the mold chills first. In doing so, it contracts, and if left to itself, might easily form contraction cavities and thin spots. If, however, the rest of the metal is still in a liquid state, it is projected, with a pressure at the rate of, say, 100 lb. per lb. of metal, against the chilled layer, and fills all possible cavities with an efficiency many times greater than can be obtained with sink heads in stationary casting. (To obtain the same results in a stationary mold, it would have been necessary to put into the sink head 99 lb. of metal for every pound of metal in the actual casting, a proposition that is not likely to appeal to a sane foundryman.) This again presupposes, however, that the cooling proceeds at a fairly slow rate, and enough time is available for each succeeding layer to take care of chill conditions in the preceding layer. In cold-mold centrifugal casting, which means with a mold warmed only enough to take off the chill, say 300 to

400 deg. fahr., temperature conditions do not favor slow cooling, which is one of the reasons of the many past failures to produce thin-walled steel tubing by centrifugal casting. On the other hand, very excellent steel tubes (both plain carbon and nickel alloy) have been produced with diameters ranging from 10 in. up, and wall thicknesses up to $3\frac{1}{2}$ in. In the latter case there is so much metal in comparison with the area in contact with the mold that enough time is available to produce a good segregation of impurities inward before the inner wall freezes.

20 It would appear, therefore, that the secret of producing centrifugally thin metal castings (under $\frac{1}{2}$ in.) lies in establishing conditions under which the cooling of the molten metal will proceed at a fairly slow rate. This applies, of course, only to such metals as steels and monel, but not to bronze, which is governed by different conditions. In the Cammen process this has been accomplished by preheating the mold to a temperature close to the melting point of the metal itself, which may vary from, say, 1600 to 2000 deg. fahr. for steel and monel-metal castings (Cp. Appendix No. 4). Under these conditions a $\frac{3}{16}$-in. wall takes about 45 to 60 sec. to harden entirely, which is sufficient to produce clean metal. Casting in such extremely hot molds is a rather novel procedure and at first one might anticipate trouble due to oxidation and warping of the molds and the difficulty of handling big molds conveniently. As a matter of fact, however, with proper equipment it offers comparatively few difficulties. The question of oxidation of molds is taken care of by using proper alloys, such as alloys of a nickel-chrome base or cast tungsten (the latter is not yet being done on a commercial scale). The question of warping is partly taken care of by the same use of proper alloys, but mainly by the use of extremely substantial molds and proper facilities for handling them between the casting machine, the transfer table and the furnace. In all this handling, the molds have to be properly supported throughout the entire length, either by carrying them on a very heavy " horn " extending from one end of the mold to the other, or in a cradle supporting the mold throughout its length.

21 The method of operation is as follows: The mold with its casting inside, both at a temperature well above white heat, are rapidly pulled out from the casting machine and carried over to what is known as a transfer table, where the casting is pushed out of the mold by a hydraulic plunger. While the casting is going to

the proper rolls or benches, the mold itself is coated inside by a protective layer and carried over to a reheating furnace. The furnaces used in this connection do not differ essentially from the billet-heating furnaces in tube mills. The mold coating is intended to protect the metal of the mold from coming into direct contact with the molten metal of the casting and need not be more than, say, $\frac{1}{64}$ to $\frac{1}{32}$ in. thick. It consists of some refractory material like alumina, zirconia, kaolin, electrically calcined magnesia, or the like. Either pitch with some addition of solvent naphtha or clay may be used as a binder, the former being preferable as it contains no water. The coating may be applied either by a swab or through a compressed-air gun.

22 The transfer table referred to above cannot be described here owing to lack of space. It is a somewhat cumbersome piece of apparatus which may weigh from 2 to 10 tons, depending on the sizes of the castings handled, and is a practically self-contained unit operated from a cab by pulling a series of levers one after another.

FIELD OF APPLICATION OF HOT-MOLD CENTRIFUGAL CASTING

23 The particular feature that determines the field of application of hot mold centrifugal casting is the ability to produce cast shapes of comparatively thin metal (down to $\frac{1}{8}$ of an inch) in great length, with our present knowledge of the art up to 20 ft. One of the first applications that comes to mind is in the production of seamless tubing.

24 Hitherto the principal tonnage of seamless tubing has been made by the Mannesmann piercing process and its variations. Essentially this process consists in imparting such a twisting motion to a steel billet as to break down its central fibers. This motion is set up by means of obliquely placed rolls or disks. There are several objections to this process. In the first place, while it does pierce high-grade mild steel or Muntz-metal billets, it stresses the central part of the billet far beyond its elastic limit. While apparently this does not affect very seriously the tensile strength of the finished product after it has been cold-drawn or hot-rolled, there is good reason to believe that it impairs the resistance of the metal to corrosion, the latter, as we know now, being particularly active in the case of metals stressed up to or beyond the elastic limit. Furthermore, the Mannesmann process is not at all applicable to

metals which are either extremely tough or very brittle. Thus, all attempts to produce commercially by the Mannesmann process, hollow billets of either monel metal or Admiralty brass have so far failed. Not only that, but the Mannesmann process, by its very character, is limited to the production of comparatively small sizes, not in excess of 6 in. in diameter; while experimentally larger sizes have been made, the 6-in. limit has been pretty well established for standard commercial grades. It is obvious that as we go up in sizes, the piercing mills used in this process increase in cost with great rapidity, and as the cost of even a small Mannesmann mill easily runs into six figures, the cost of the big ones becomes prohibitive after passing a certain limit, which is around 6 in.

25 At the same time there is a demand for larger sizes of tubing of greater strength than a single-welded joint can give and this is where the most obvious field of hot-mold centrifugal casting appears. With this process, tubing in extra heavy and double extra heavy thicknesses can be easily produced in sizes up to 24 in. in diameter at a cost per ton that can compete with plain welded tubing and it is considerably lower than that of double-welded tubing such as is used for hydroelectric installations.

26 The next field of application in connection with steel tubing is in standard sizes from 6 to 14 in. up, where welded tubing is now used exclusively. Such pipe can be produced in two ways: First, by casting a hollow billet of wall thickness two to three times that of the finished pipe, and either cold-drawing or hot-rolling it to size. In such a case the billets cast would be roughly 8 to 10 ft. long so as to have say, 22 ft. in the finished pipe, barring cropped ends. The other method is to cast the pipe direct with only a slight excess of wall thickness and then to give it one pass between straightening rolls over a ball, so as to reduce the diameter and wall thickness to exact size with standard tolerances, and give the exterior and interior walls a finished appearance. From data now available it would appear that the final cost of both kinds is approximately the same and is well within the range at which centrifugally cast-steel pipe can compete in price with welded pipe.

27 It is only natural that this should be so. In pipe manufactured by the standard processes, steel is cast in ingots, which, after cropping and several thermal and mechanical operations, are rolled into skelp. The skelp goes to a pipe mill, is heated to a red heat and rolled to bevel edges, after which it is sent through the bending rolls. It is next reheated to welding heats and in succes-

sion passed through welding rolls, sizing rolls and finally straightening rolls. Each of these operations is done, of course, by appropriate machinery developed to a remarkably high state of perfection, but nevertheless each operation adds its mite to the final cost of the article, and this mite is not inconsiderable in view of the immense tonnage produced. In centrifugal casting the metal is poured into the mold which corresponds to casting the ingot, and the casting is then ready to go to the final rolls, which can be done without reheating. There is at least 50 per cent less handling of material in centrifugal casting than there is in making pipe by welding, and such a difference cannot help being reflected in costs.

28 In this connection it might be well to mention one of the psychological factors with which centrifugal casting is concerned, and that is, the not unfounded distrust of users of metal products toward castings, especially those in thin sections. Such castings cannot be produced commercially in a stationary mold, and steel castings in a stationary mold as made today generally are of such physical properties as to be usable only where they are not subject to complicated stresses and where an extremely high factor of safety can be provided. Wherever the stresses are high and especially uncertain and the factor of safety is not more than ample, no engineer today will use a casting but will insist on either a forging or a piece machined or hot-rolled.

29 This not unfounded distrust of castings is due to the fact that in either the sand casting or chill casting the temperature conditions in the metal are uncontrollable, with the result that one part of the metal may be entirely different in its physical properties, such as hardness and crystalline structure, from the other, and there is no guarantee that two castings made under apparently the same conditions will be entirely alike. There is too much individuality in products cast in stationary molds to make them suitable for engineering requirements where a failure would be disastrous and where only moderate factors of safety may be employed. The situation is, however, entirely different with centrifugal castings, because there ample facilities are available, especially in hot-mold casting, to control the rate of cooling of the metal, and the nature of the process is such as to tend to give a uniform product, provided, of course, the process is carried out properly. From this point of view, one cannot help agreeing with an editorial in the *Iron Age* (Feb. 9, 1922), which states that "the successful operation of centrifugal casting processes insures a product, no mat-

ter of what composition, which is of a high grade. It is a realization of quality production in quantity, for rapid output is a marked characteristic. There is also the advantage of the elimination of sand and dirt. The condition of casting and cooling tends to produce a dense casting and one whose microstructure is different from the sand-cast products. Centrifugal force is substituted for sink heads as an insurance against unsoundness." This brief analysis in the *Iron Age* states strikingly the difference between sand castings and centrifugal castings, and explains the reason why centrifugal castings may be used on a par with forged or hot- and cold-worked metal where sand castings would be too hazardous.

30 But the field of application of centrifugal casting, especially of the hot-mold type, does not stop with plain carbon steels. There are a number of alloy steels which it would be very desirable to have in tube form, such as stainless steel, high-speed tool steel, chrome-nickel-vanadium steel and the various " near-steels " which are iron alloys with a predominance of materials other than iron, such as heat-resisting alloys. Practically none of these materials will stand for piercing by the Mannesmann process, but cast with great ease into tubular shapes. In fact, even Hadfield manganese steel has been found to make, by centrifugal casting, excellent tubing, although it is very doubtful if it has any commercial application outside of some very limited specialties.

PLATE MANUFACTURE

31 Centrifugal casting is peculiarly applicable to the manufacture of plates. Andrew Shanks, the British inventor of the middle of the 19th century, to whose pioneering work in the manufacture of cast-iron pipe reference has been made above, was also the earliest manufacturer of plate. His process was to cast a thin-walled pipe, cut it, and then by careful annealing and hammering, flatten it out. The process was kept secret and was successful commercially until rolled steel sheet was put on the market.

32 The use of a hot mold makes it possible, however, to secure comparatively thin-walled castings in large sizes. A process has been worked out, though not yet applied on a commercial scale, in which a cylindrical casting is made in such a manner that a longitudinal split of about half an inch wide is produced. The split cylinder as it comes from the mold is sent first through a flattening jig where it is flattened out on its own heat, and then to rolls, of which there are at least two, one for sizing and the other for finishing. Under

certain conditions more than one sizing roll may be required. The casting has to be cast oversize, the wall thickness being from 10 to 15 per cent greater than that of the finished plate. Thus, for example, one-quarter-inch plate would be cast in sizes 240 in. long, 20 in. in diameter and 0.265 wall thickness. It is expected that it can be brought down to a quarter of an inch in one pass in the sizing rolls and that no reheating would be necessary.

33 This process is today only in its initial stages of development. It is attractive, however, as it does away with the ingot-casting work, soaking pits, blooming mill and a great share of the rolling equipment, permitting a conversion from molten metal to finished plate estimated at about $4.00 per ton.

34 Not only that, but it also makes quite attractive the manufacture of alloy-steel plates, in particular, of such material as acid-resisting steels for use in chemical tanks and the like. These steels are very difficult to roll. The final product could stand the cost of the two or three passes necessary in centrifugal casting of plates, but not the many operations of the conventional methods of plate manufacture, the main item of expense being not the actual rolling but the complicated heat treatment absolutely imperative between the rolling operations. In fact, it would not be surprising if with prices of alloy-steel plates brought within reason, they would be used far more extensively than one would expect to find from present indications.

35 The casting of gears by centrifugal methods, in particular worm and herringbone gears, has been achieved with great success both in England and in America, especially in this country. As far as the writer is aware, however, all such gears have been cast in non-ferrous metals and not in steel.

36 There is one more field of application of centrifugal casting, especially the hot-mold type, comparatively small today, but which may become of material interest shortly, and that is the production of cylinders requiring extremely high strength. When the oxygen industry first called for cylinders that would safely withstand 2000 lb. pressure to the square inch, it looked like a big problem, but it was solved by the progress in the cupping process. When, however, the Haber synthetic ammonia process demanded great pressures combined with considerable temperatures, special expensive tools had at first to be provided for machining the cylinders required from solid stock. The Claude ammonia process goes to temperatures and pressures far exceeding those of

Haber, and the experiments of Professor Bridgman of Harvard University and others point to the likelihood that before many years are past pressures of the range of 100,000, 200,000 and possibly even more pounds to the square inch will be used commercially. At pressures between those used by Haber and Claude and those used by Professor Bridgman for containers of commercial size, carbon steels are but little suitable and the vessels required will have to be made from alloy steels.

37 This may be done in two ways, either directly by centrifugal casting on a vertical axis which would give a vessel with closed bottom, or by the cupping process from alloy-steel plates, which, in their turn, would be made by centrifugal casting as described above. In this connection it may be of interest to mention that a high measure of success was achieved during the war by the Bethlehem Steel Company at Bethlehem, Pa., in casting air flasks for torpedoes centrifugally in a machine running on a vertical axis.

38 No attempt has been made in the above discussion of the field of application of centrifugal casting to present anything like an exhaustive list. Instances have been merely cited to show the enormous field in which this process is or may be employed.

39 The same applies largely to the description of the mechanics of the process, where likewise no attempt has been made to show the historical development of machinery for centrifugal casting or to give details of the actual apparatus. The apparatus has been shown mainly in order to give an idea of the kind of machinery that has to be used.

40 In connection with the mechanics of centrifugal processes it seems appropriate to remark at the close of the paper so as to give it the greatest emphasis possible, that the machinery itself is and should be comparatively simple, for the process when properly worked out is the acme of simplicity. It should be, however, clearly understood that in centrifugal casting we are dealing with large rotating masses and with metal subjected in its molten state to the effect of great force.

41 Furthermore, the distribution of the metal in the mold is effected primarily not by rigid material visibly distributed as in stationary casting, but by the action of invisible forces which do not come into operation until the moment when the metal is delivered to the mold.

42 Because of all this, and notwithstanding the great sim-

plicity of the machinery and methods of centrifugal casting, it should be clearly remembered that even the slightest imperfection in the design or operation will immediately show up in the casting.

43 Centrifugal casting is a process which is peculiarly impossible to be worked either by slipshod methods or on a shoestring, i.e., skimping on the quality of materials and on factors of safety in the design of machinery.

44 There is an old saying current among molders to the effect that a lie in sand will be shown up in the metal. The same applies with still greater force to centrifugal casting. The slightest mistake in the layout of the machinery or use of poor materials in the machine, its bearings and foundations, will produce an uneven casting, result in excessively rapid wear of the molds, and at times may even cause disastrous accidents with danger to life. At the same time, with good engineering and the use of first-class materials, centrifugal casting may be carried on entirely by semi-skilled labor and still give products of unsurpassed excellence.

APPENDIX No. 1

BIBLIOGRAPHY

45 The articles enumerated below contain references to previous publications on centrifugal casting: These, as a rule, have not been included.

1 Radial Stresses Due to Centrifugal Force in Revolving Bodies, A. M. Levin. *American Machinist*, vol. 27, no. 43, Oct. 20, 1904, pp. 1389–1393, 6 figs.

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10 Tests of Centrifugally Cast Steel, Dr. Geo. K. Burgess, *Transactions of the American Society for Steel Treating*, vol. 1, no. 7, April, 1921, pp. 370–382 (published also as a Technical Paper of the Bureau of Standards, and abstracted in *Iron Age*, vol. 107, no. 12, March 24, 1921, pp. 764–766; cp. editorial in same issue, p. 789).

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13 Casting Rings in Centrifugal Machine, E. F. Cone, *Iron Age*, vol. 102, no. 14, October 3, 1918, pp. 801–807.

14 Centrifugal Castings (The Stokes machines), *Engineering*, vol. 111, no. 2881, March 18, 1921, pp. 311–312.

15 English Make Centrifugal Castings, H. Cole Estep (same as the preceding item, but more complete), *Foundry*, vol. 50, no. 6, March 15, 1922, pp. 217–222.

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17 Mold Serves as Electric Furnace, E. C. Kreutzberg, *Iron Trade Review*, vol. 49, no. 1, July 7, 1921, pp. 33–35 (description of the electric type of the Cammen process, reprinted in *Foundry*, vol. 49, no. 19. October 1, 1921, pp. 772–774).

18 Steel Pipe by the Centrifugal Process, L. Cammen, *Iron Age*, vol. 109, no. 6, February 9, 1922, pp. 405–406 cp. editorial in the same issue, Progress in Centrifugal Casting, p. 426).

19 Centrifugal Casting, L. Cammen, *Chemical and Metallurgical Engineering*, vol. 26, no. 8, February 22, 1922, pp. 354–358.

20 Ledebur, Handbuch der Eisen- und Stahlgiesserei, 1892.

21 B. Osann, Eisen- und Stahlgiesserei, 1920, pp. 423–424.

46 The following references apply to special processes:

Huth: 7, 21
DeLavaud: 11, 12, 13
Stokes: 2, 14, 15
Millspaugh: 10
Cammen: 17, 18, 19
Hume: 8

APPENDIX No. 2

U. S. PATENTS ON CENTRIFUGAL CASTING

47 The following list gives the numbers (which are sufficient to locate the patents) of patents issued by the U. S. Patent Office on centrifugal casting or machines used thereon. So far as a careful search through the records of the Patent Office has been able to establish, these are all that have been issued; in any event, the compiler is confident that no important patents dealing with the art have been omitted. Those desiring to secure copies of the patents should bear in mind that the majority of the earlier patents are out of print and can be obtained only by ordering photostats. Copies of patents which are available can be obtained by writing to the Commissioner of Patents at the regular rate of 10 cents per copy.

1,403,347	1,386,268	1,410,885	1,377,226
1,377,406	1,385,781	1,376,090	1,358,435
1,377,372	1,362,821	1,360,484	1,352,523
1,366,875	1,349,416	1,351,766	1,344,409
1,351,767	1,329,295	1,347,582	1,320,744
1,340,337	1,329,179	1,323,252	1,331,660
1,329,296	1,325,039	1,329,754	1,310,865
1,302,584	1,323,253	1,324,660	1,307,823
1,229,918	1,304,118	1,309,823	1,302,838
1,315,512	1,302,448	1,320,910	1,286,555
1,300,123	1,277,543	1,293,049	1,263,208
1,283,139	1,215,243	1,280,289	1,223,676
1,276,038	1,240,546	1,239,994	1,199,281
1,257,325	1,169,192	1,199,353	1,195,803
1,177,957	1,161,701	1,190,009	1,142,840
1,128,009	1,124,099	1,150,974	1,113,333
1,124,841	1,110,954	1,121,659	1,095,523
1,113,017	1,066,604	1,109,572	1,058,250
1,095,230	1,028,330	1,080,224	1,040,517
1,047,972	1,015,818	1,004,293	1,009,167
1,015,819	986,130	950,106	985,489
964,371	951,614	971,957	930,828
914,459	918,880	893,037	839,861
839,560	777,560	767,928	727,103
655,235	652,756	734,375	743,077
644,926	648,601	644,018	644,019
581,272	550,089	538,835	522,093
510,995	482,405	467,469	462,423
359,484	462,422	444,162	427,708
422,590	412,687	368,746	359,483
348,709	224,245	110,717	104,130
96,911	65,642	48,022	98,673
35,884	24,915	24,539	18,924
5,988			

APPENDIX No. 3

SOME MATHEMATICAL CONSIDERATIONS ON THE SUBJECT OF CENTRIFUGAL CASTING

48 This is not intended to be a complete treatment of the complicated phenomena taking place in centrifugal casting, but merely a somewhat elementary presentation of some of the more important facts that will help in understanding the processes and in estimating the forces acting therein.

49 Centrifugal casting may be carried on with the axis of rotation horizontal, vertical, or inclined, although one may consider the case of casting about the inclined axis as the general case, and the horizontal and vertical positions as special cases at the two limiting extremes. Finally, there is the case, of no apparent practical importance, of casting about axes of double rotation, that is, where, in addition to the rotation of the mold, the entire machine is spun about some axis which may and may not be external to the mold. Practically, such a case would arise if, in addition to spinning the mold, the entire machine were placed on a turntable and given a rapid rotation, say, about some vertical axis. This last case will not be considered here.

CASTING ABOUT A VERTICAL AXIS

50 The case of casting about a vertical axis is the simplest theoretically, and of considerable commercial importance, as nearly all castings of gears, railroad wheels, etc., in which already quite a large production has been built up, are made on a vertical axis.

51 There are several ways of proving that a liquid subjected to rapid rotation about a vertical axis will assume on its inner surface the shape of a paraboloid. Horace Lamb [1] proves it by deriving the common integral of the dynamical equations of the rotating liquid. A simpler proof is offered by Prof. V. L. Kirpicheff.[2]

52 For a particle of liquid m, Fig. 4, on the inner surface of the body formed by the centrifugal force, determine the force mc considered as a resultant of the force of gravity mb and the centrifugal force ma which in its turn is equal to mq^2r where q is the angular velocity of rotation of the vessel.

53 The laws of equilibrium of liquids demand that the surface of the liquid at m be normal to the direction of the force mc, which means that mc must be normal to the curve $AmBC$ at m. Continue this normal until it intersects the axis of rotation, say, at d. Since the triangles mdo and mbc are similar, we have

$$\frac{od}{om} = \frac{mb}{bc} = \frac{mg}{mq^2r}$$

Hence

$$od = \frac{g}{q^2} \quad \ldots\ldots\ldots\ldots\ldots \quad [1]$$

[1] Hydrodynamics, 1906, p. 24.

[2] Talks on Mechanics, St. Petersburg, 1907, pp. 94–95.

which proves that the subnormal *od* to the curve *AmBC* does not depend on *r*, and consequently is the same for all points of this curve. But the subnormal is constant only for a parabola. Hence, *AmBC* is a parabola, and the "funnel" formed by a rotating liquid is a paraboloid of revolution.

54 Lamb (*loc. cit.*, Par. 51) proves also that in a mass of liquid rotating about a vertical axis with constant and uniform angular velocity and only under the action of gravity a velocity-potential cannot exist, from which it follows that a motion of this kind could not be generated in a "perfect" fluid, i.e., in one unable to sustain tangential stress.

55 Hans Lorenz [1] offers a carefully-worked-out proof to the effect that the distance to which the apex of the paraboloid of revolution of the liquid mass

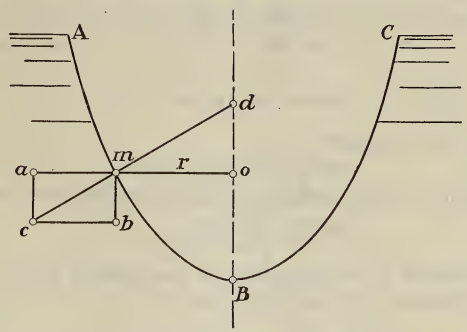

FIG. 4 FORCES ACTING IN A LIQUID IN A VESSEL ROTATING ABOUT A VERTICAL AXIS

rotated about a vertical axis, as referred to above, is depressed, is proportional to the square of the peripheral speed of rotation.

56 The complete derivation of the internal shape of a liquid rotating presumably at a considerable speed about an axis inclined at an angle, α, to the vertical (that is, the direction of acceleration due to gravity) is given by Hans Lorenz, who starts with the equations of motion of the liquid, together with the equation of continuity

$$\frac{\partial(w_r r)}{\partial r} + \frac{\partial(w_z r)}{z} = \frac{\partial w_n}{\partial \phi} = 0 \quad [2]$$

57 He proceeds to make certain substitutions in the equations of motion referred to above. The integration of the equations remains, however, extremely difficult if not impossible. Lorenz therefore assumed that it would be permissible to neglect the axial and radial components and retain only the tangential velocity

$$w_n = \epsilon r \quad [3]$$

where ϵ is the constant angular velocity of rotation.

58 From Fig. 5 he derives

$$q_r^2 + q_n^2 + q_z^2 = g \quad [4]$$

[1] Technische Hydromechanik, 1910, p. 372.

and this together with Equation [3], which, by the way, satisfies the equation of continuity [2], permits him to write

$$\left.\begin{aligned} g \sin \alpha \cos \phi + \epsilon^2 r &= \frac{g}{\gamma} \frac{\partial p}{\partial r} \\ -g \sin \alpha \sin \phi &= \frac{g}{\gamma} \frac{\partial p}{r \partial \phi} \\ g \cos \alpha &= \frac{g}{\gamma} \frac{\partial p}{\partial z} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [5]$$

59 These Equations are consecutively multiplied by ∂r, $r\partial\phi$, ∂z, added together and integrated, which gives

$$gr \sin \alpha \cos \phi + gz \cos \alpha + \frac{\epsilon^2 r^2}{2} = \frac{g}{\gamma} p + C \quad \ldots\ldots\ldots [6]$$

Next, there is added to both sides the member

$$\frac{g^2 \sin^2}{2\epsilon^2} = \frac{a^2 \epsilon^2}{2}$$

which gives

$$\frac{\epsilon^2}{2}(r^2 + a^2 + 2ar \cos \phi) + gz \cos \alpha = \frac{g}{\gamma} p + C_1. \quad \ldots\ldots [7]$$

60 Further, the expression

$$r^2 + a^2 + 2ar \cos \phi = \rho^2 \quad \ldots\ldots\ldots\ldots [8]$$

defines the direction ρ of an element of liquid located at a distance

$$a = \frac{g \sin \alpha}{\epsilon^2}. \quad \ldots\ldots\ldots\ldots\ldots [9]$$

from a new axis parallel to the original axis but located above it. Because of this, instead of Equation [7], we may write

$$\frac{\epsilon^2 \rho^2}{2} + gz \cos \alpha = \frac{g}{\gamma} p + C_1. \quad \ldots\ldots\ldots\ldots [10]$$

61 If this equation be considered with respect to the variables ρ and z at constant pressure, it will be seen that it describes a paraboloid of revolution differing from similar paraboloids of revolution defining the motion of a liquid about a vertical axis, in that in the member containing z we find $g \cos \alpha$ instead of g. Because of this, the distance of the apex of the paraboloid from its topmost edge z_1 is

$$z_1 = \epsilon r_0 \sqrt{\frac{h_1}{g \cos \alpha}} \quad \ldots\ldots\ldots\ldots [11]$$

as compared with the distance between the same elements in the case of a rotation about the vertical axis,

$$z_1 = \epsilon r_0 \sqrt{\frac{h_1}{g}} \quad \ldots\ldots\ldots\ldots\ldots [12]$$

62 A comparison of these equations would indicate that in the case of an inclined axis z_1 increases very rapidly with the speed as the angle of inclination of the axis increases. This leads at once to the derivation of the basic formula for the case of spinning about the horizontal axis:

$$\frac{\epsilon \rho^2}{2} = \frac{g}{\gamma} p + C_1 \quad \ldots\ldots\ldots\ldots [13]$$

in accordance with which the surfaces of equal pressure in a liquid rotating about a horizontal axis are of approximately cylindrical shape and their common axis is at a distance, a, over the axis of rotation, a being defined by equation

$$a = \frac{g}{\epsilon^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

CASTING ABOUT A HORIZONTAL AXIS

63 The mathematics of casting about a horizontal axis are essentially extremely simple. The liquid acts practically like the rim of an ideal flywheel, that is, one which either has no arms or arms capable of infinite extension without effecting any pull on the rim.

64 Three forces are acting on the liquid in a horizontally located mold: namely, gravity, friction of the liquid against the wall, and centrifugal force. If there were no friction of the liquid against the wall, the liquid would not be subject to the centrifugal force and the mold would simply spin around it while the liquid would retain its level. It is important to realize this fact, although numerically hitherto no way has been worked out to take into consideration the slip of the liquid except by providing for it in determining the spinning speed. The subject is extremely obscure as yet, because of a total lack of experimental data. Assuming, therefore, that the coefficient of friction between the liquid

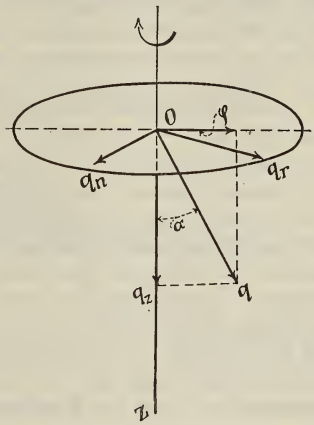

FIG. 5 FORCES ACTING ON A LIQUID IN A VESSEL ROTATING ABOUT AN INCLINED AXIS

and the walls of the mold is such that the liquid moves at the same peripheral speed as the walls of the mold in contact therewith, one may apply to the calculation of the centrifugal speed acting on the liquid the same formula as is applied to the rim of a flywheel, namely,

$$F = 0.000341\, WRN^2,$$

where F is the total centrifugal force in pounds, W the weight of the liquid in pounds, R the radius in feet, and N revolutions per minute. Where the thickness of wall of the cast article is small, it is advisable to take the inside radius as R. Where the thickness of the wall is considerable, the mean radius may be taken by adding the inside and outside radius and dividing the sum by two. Finally, where the thickness of the wall is very large and the radius small, it is well to use for R the value of the outside radius, which is, strictly speaking, erroneous, but permissible, as the error lies on the safe side. As a rule, however, the mean radius will give values precise enough for all practical purposes.

65 Experiments have shown that for some metals, certain speeds of rotation have to be maintained, and it would appear that the speeds of rotation that have to be maintained depend partly on the temperature range between the pouring temperature of the metal, its freezing temperature and the temperature of the mold. The speed of rotation also depends to a certain extent on the thickness of wall which has to be cast. Experimentally it has been found that for steel the speed may be expressed as $N = \frac{1675}{\sqrt{r}}$ where r is the inside radius of the casting in inches. If this value be substituted in the equation for centrifugal stress, it will show that each pound of metal in the casting exerts a centrifugal pressure equal to 75 lb.

66 Essentially, centrifugal pressure does not depend on the specific weight of the metal, a pound of water at a given speed exerting the same centrifugal force as a pound of mercury. Actually, however, it is not quite so, because the value of centrifugal force depends, as seen from the above equation, on R as well as on W. Now, a pound of water occupies 13.6 times as much space as a pound of mercury, which affects the value of the mean radius whether derived by approximation or by formal mathematical methods.

FORCES ACTING ON THE LIQUID IN A HORIZONTAL MOLD IN THE DIRECTION OF THE AXIS OF THE MOLD

67 From a practical point of view this is an extremely important subject which, as far as the author is aware, has not been yet given due consideration. The importance of this subject lies in the fact that it is the forces acting in a longitudinal direction that permit the metal poured in at one end of the mold to be spread the entire length of the mold, and this in turn permits using a short stationary spout, avoids the use of such complicated and at times unsatisfactory devices as trough spouts, sliding spouts, etc.

68 In order to bring the problem to its simplest elements, it is assumed that the mold is of considerable length, say, 20 ft., of small diameter, say, 4 or 5 in., and is at room temperature, and that the liquid therein is mercury. It is further assumed that the speed at which the mold is spun is such as to satisfy the above equation in such a manner that $F = 75$ lb. when $W = 1$ lb.

69 In Fig. 6A, m is the mold; assume now that while the mold is spinning, a batch of mercury has been poured in so as to form a ring w, 3 in. long and $\frac{1}{2}$ in. in wall thickness; assume further that by some means such a ring has actually been produced, if only for an instant, and that, as the ring begins to spread, new metal is supplied to maintain its shape.

70 Take the droplet of mercury P, Fig. 6B, which shows part of Fig. 6A in an enlarged form. It is acted on by its own centrifugal force equal to 100 F, where F is its weight. Furthermore, it undergoes a push on the part of the small cylinder a, equal roughly to 100 A, where A is the weight of the cylinder a. Because of this, the droplet P exerts a pressure on all the droplets around it, and in particular tries to push the droplet C immediately between it and the wall of the spinning vessel and thus displace it in the direction toward the wall. Droplet C does the same thing to the next one back of it, and this process continues until we reach droplet D, which is right next to the wall. This droplet, according to the elementary laws of hydraulics, exerts a pressure which is the same in all directions. The pressure exerted against the wall of the spinning

mold produces in it a stress, but the wall is supposed to be solid enough to resist this stress without appreciable deformation.

71 On the other hand, however, droplet D exerts a pressure on droplet E, and as there is nothing to hold droplet E in its position except friction, which is comparatively slight in this case, droplet E moves away in the direction e. This leaves droplet D unsupported on one side, and as droplet G exerts on it the same kind of pressure that it does on E, D moves into the place of E. In this way there is a continual displacement of the droplets in both directions until they encounter a solid wall and a state of rotational equilibrium has been established, which means until a uniform cylinder has been formed from one end of the mold to the other and the supply of new metal has ceased.

72 Let us see now what are the forces that propel the metal and force it to spread in the manner indicated. Assume that the mold is 6 in. inside diameter and that molten steel is poured from the spout at such a rate as to form the ring w $\frac{1}{2}$ in. wall thickness and only $\frac{1}{2}$ in. long, instead of 3 in. as shown in Fig. 6. The weight of the metal in the ring is then slightly over 1 lb. per sq. in. of the area of the mold wall covered by metal, and the pressure per sq. in. of

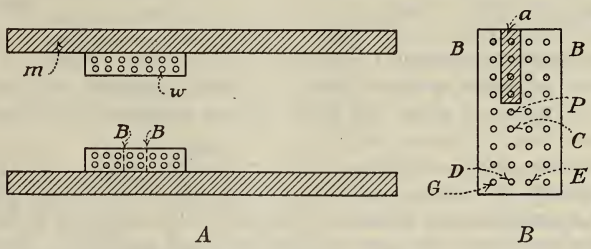

FIG. 6 FORCES CAUSING THE LIQUID IN A VESSEL ROTATING ABOUT A HORIZONTAL AXIS TO SPREAD LONGITUDINALLY

(A, left, mold and ring of liquid; B, right, a part of the ring of liquid in A, magnified)

wall, or which is equivalent to it, per sq. in. of molten metal, is 100 lb. At such a pressure molten steel, having to overcome merely its own friction and the friction against the walls of the molds, ought to make the distance of 20 ft. in something under $\frac{1}{10}$ second.

WHAT PREVENTS THE METAL FROM SPREADING IN A COLD MOLD?

73 Great difficulty has been experienced in getting the metal to spread in a cold or only slightly heated mold, which led to the development of elaborate sliding spouts, trough spouts, and the like. The reason for this difficulty lies in the fact that the above reasoning is correct only for the case of liquids, such as mercury or molten and perfectly fluid metals, as it is only in liquids that the pressure at a point is equal in all directions.

74 But the metal in a cold mold does not stay fully liquid for any appreciable length of time. As the particle D is projected beyond the ring w, it strikes the cold wall of the mold. The metal film is at that instant so thin that the amount of reserve heat in it is very slight. When this thin film of molten metal comes in contact with the mold, which is say, 2000 deg. colder than the metal, and many times its volume, the metal freezes, and in doing so forms a hard and non-liquid fringe about the ring w. This at once tremendously increases

the frictional resistance of the spread of the metal, and in fact creates about the same situation as a dam of rocks in a river bed. The flow is impeded, in addition to which chilling of the metal is assisted by the slower flow.

IS CENTRIFUGAL FORCE A REAL FORCE OR A MATHEMATICAL FICTION ?

75 This question has been asked several times and is due to the fact that, notwithstanding its great importance in mechanical engineering, the subject of centrifugal force in our textbooks of physics is considered in a very cursory and superficial manner. On the one hand, every engineer knows that there is some force that produces effects ascribed to what is called centrifugal force. On the other hand, however, an incorrectly understood theory of the action of this force makes the subject a good deal more mysterious than it needs to be. Here is how the matter appears from a cursory reading of our textbooks on physics. Let us assume that a ball attached to a string is whirled about a fixed center. Then, the mass m of the ball moves in a circle owing to the tension in the string which pulls the ball toward the center. This is the centripetal force, the magnitude of which is equal to the mass m times the acceleration, or $\frac{mv^2}{r}$, where v is the uniform circular speed and r the radius of the circle. Because of the action of this force the initial motion instead of being rectilinear becomes circular. Thus far everything is clear, but from now on a misunderstanding is apt to arise.

76 Assume that in addition to the centripetal force, a centrifugal force acts on the body m and our textbooks usually state that the centrifugal force is equal in magnitude and opposite in direction to the centripetal force. Since however, the two forces are equal in magnitude and opposite in direction they ought to cancel each other, so that if it be assumed under the above reasoning that the centrifugal force is a real force and not a mathematical fiction, it would mean that no force is acting on the ball as the two forces, which we have assumed to be active, mutually cancel. But if that were the case, there would have been no reason why the ball should not follow the rectilinear motion, and yet, instead of doing that, it moves in a circle. There is, therefore, a contradiction apparently created by the assumption that centrifugal force is a real force.

77 The error in this reasoning lies in the imperfect understanding of the action of the two forces and more precisely in the erroneous assumption that both forces, — the centripetal and centrifugal, — act on the same body, namely, the ball.

78 Actually, however, we have here a system consisting of *two* bodies — the ball and the string. The action of the string on the body is due to centripetal force, while the reaction of the body on the string is due to the centrifugal force, the action and reaction here as elsewhere being equal and opposite in accordance with the third law of Newton.

79 Centrifugal force is therefore a perfectly real force and may be measured by a dynamometer placed in the string. The centripetal force is also a perfectly real force, though there does not appear as yet to be any way of directly measuring it, nor is any necessary as it can be determined by measuring the centrifugal force and changing the sign of direction.

APPENDIX No. 4

TEMPERATURE CONTROL IN CASTING AND MOLD

80 The purpose of temperature control of the metal of the casting and of the mold is to give time to the metal to get rid of its occluded gases and slag, which, *if time is available*, is powerfully assisted by the action of centrifugal force, and also to give time to supply hot metal to contraction cavities formed during the early stages of the cooling of the metal. This has been more fully explained in Pars. 16–20. The matter is of such importance that it is no exaggeration to say that if, in centrifugal casting, you take care of the mold, the metal will take care of itself.

81 The above brief exposition of the purposes of temperature control in centrifugal casting provides us at once with a criterion as to the proper temperature of the mold at the time of pouring the metal. We have to give time to the metal to perform certain functions which it can perform only while it is still liquid, and this means that the mold must be at such a temperature that the metal of a given casting, while in contact with the mold, should remain fluid for the necessary length of time. This, in formal language, may be expressed as follows: The temperature of the mold should be such that the loss of heat from the metal to the mold at its obtaining temperature be less than that which is sufficient to chill the metal to the point of congelation within the time period necessary for the formation of the proper casting shape and for the escape of the gases from the metal.

82 The following calculation may help to illustrate the application of the above broad principle. It is assumed that a metal mold is used to make a steel pipe, say, 6 in. in diameter and of standard thickness (0.280 in. wall, say, 20 lb. per ft.). Experimentally it has been established that for casting such a size, the metal should remain fluid after pouring for a period of not less than, say, 30 sec. Assume that the metal, when coming out of the ladle, has a temperature of, say, 2800 deg. fahr., which is probably higher than it will actually have in an open-hearth plant. Even with this temperature, the metal has, between the liquid and solid states, a reserve range of only 300 deg., equivalent to about 650 B.t.u., neglecting latent heat.

83 The reserve heat is here so low that if a mold of conventional type were used at a temperature of, say, 400 deg. fahr., the loss of heat to the mold would have been so rapid that the metal of the casting would freeze in not more than a couple of seconds, as has been proved by actual experiment. An extremely hot mold therefore becomes necessary, and tests have shown that to obtain good castings, one must have the mold at bright white heat, or 1800 to 2000 deg. fahr. Furthermore, it is desirable not to have the mold made of a metal that is a very good conductor of heat, as the prime function of the mold is to conserve the heat in the casting, and not to carry it away.

84 On the other hand, however, where a metal mold is used, unless such metals as cast tungsten are available, which is not yet the case, there is a certain

limit to the temperature that the mold itself will stand without collapsing under the combined action of temperature and centrifugal stresses. It would appear that chrome-nickel alloys can stand temperatures as high as 2200 deg., possibly 2300 deg. fahr. for a considerable number of castings. This gives us at once a basis for calculating the size of the mold as affected by temperature conditions. (The question of stresses has also to be considered, cp. Appendix No. 5.)

85 Taking again the above example of a 6-in. standard pipe, we see that it weighs 20 lb. per foot, and is cast at a temperature of 2800 deg., while it is specified that the mold should not be heated to a temperature above 2200 deg. after starting with an initial temperature of 2000 deg. The weight of the mold must therefore be such that it could absorb all the heat in the metal down to the point of temperature equilibrium (2200 deg. fahr.) without exceeding its own limit of safety. Assume that the specific heats of the mold and metal of the casting are equal, and that the weight of the mold per foot of casting is X. Then

$$X = \frac{20(2800 - 2200)}{(2200 - 2000)} = \frac{20 \times 600}{200} = 60 \text{ lb.}$$

86 The following has to be taken into consideration before we can accept this figure definitely. The heat taken up by the mold from the casting is only partly retained by it, causing it to rise in temperature; it is partly dissipated to the surrounding air, gibs, barrel and foundations of the spinning machine (which, by the way, have to be sufficiently massive to be able to do this without becoming excessively hot). It is well, however, to neglect this factor, in determining the size of the mold, and to treat the fact of the loss of heat from the mold to the ambient medium merely as a factor of safety. Under these conditions, and taking into consideration temperature conditions alone, it is perfectly safe to make the mold of a weight equal to $1.5X$, or say 90 lb. per foot. (Actually, 100 lb. per foot has been found desirable, to take care of stresses.)

87 Suppose now that instead of casting a pipe of standard thickness, 0.280 in., it is desired to cast a cylinder with walls 1 in. thick. Such a cylinder weighs roughly 53 lb. per foot, as compared with 20 lb. for standard pipe. Accordingly, it contains more than 2.5 times as much heat above the point of solidification as did the former casting, and, because of this, does not chill quite as easily. As a result, it does not need to have the mold quite as hot as in the former case, and, as a matter of fact, a mold at 1200 to 1500 deg. fahr. will handle this kind of casting very well. The same mold as above is therefore applicable. It will be noted that in all cases a range of temperatures is indicated, rather than a single figure. This is due to the fact that in determining the temperature to which the mold has to be heated, factors in addition to those discussed above have to be taken into consideration, e.g., the length of the casting which affects the proportion of end losses to the total heat losses from the casting; the heat conductivity of the metal of the mold; the character of the material used for floating the mold, etc.

88 The underlying purpose of heat control is, however, the same in all cases, and that is, to give the metal time to clear itself of gases, and to have the freezing proceed at such a rate as to fill up contraction cavities with good molten metal. To do this, the mold has to be heated to the proper temperature, and if the mold and metal are at the proper temperature, there is no reason why a good casting should not be obtained *every time.*

89 The above does not, however, exhaust the subject of heat control, and applies only to metals which are not adversely affected by slow cooling from the liquid state. Bronze is one of the metals that are, and when such metals have to be cast centrifugally, especially in fairly thick sections ($\frac{3}{4}$ in. and up), the control of temperature has to be exercised during a much longer period than, e.g., with steel or monel metal. The first period, that is pouring and, say, 30 to 45 seconds thereafter, is governed by the same laws as have been discussed above, i.e. the temperature of the mold is held initially at such a level as to permit the metal to throw out the gases and slags, and to form the first dense layer directly against the mold wall. If, however, the same conditions were maintained throughout the rest of the cooling process, the latter would be very slow, and there is danger that the grain of the metal would be coarse and the casting unsatisfactory. It is therefore necessary to cast the metal hot and against a hot mold, but, after the tube has been formed, to chill it rapidly.

90 Some makers of centrifugally cast articles of bronze and pure copper have attempted to meet these requirements by squirting water through a hose onto the casting after it has been poured and had a chance to spin up. Good results have been obtained where the casting is short, less satisfactory with long castings. One does not have to look far for the reason of this discrepancy. In centrifugal casting there is a tendency, due to the action of the centrifugal force, to force the metal from the inner wall to the outer wall. Therefore the freezing of the metal should proceed from the outside inward, the inner molten metal supplying material for filling up contraction cavities. In short casting, the stream of water cools the mold as fast as, and possibly faster than, the metal of the casting. In long castings, however, 15 ft. and up, the water has no chance to cool the mold, with the result that the inner wall freezes first, which is contrary to what should take place in order to make a good casting. The lack of proper control of the cooling process in bronze castings leads to uncertainty of results which is possibly the explanation of the fact that some of the present manufacturers refuse to sell rough castings: not being certain as to what the state of the metal would be found to be after machining, they prefer to scrap bad castings themselves rather than have their customers do it. This may be good business policy, but indicates a rather primitive state of engineering.

91 In the Cammen process a special type of machine is used in which the mold is heated previous to casting (for common bronze, to about 1400 deg.), and then, at the proper time (which depends mainly on the size of the casting), is cooled by a stream of water. With such an arrangement, the cooling of the casting proceeds from the outer walls inward, as it should, and means are available to insure uniformity of results. Iron is probably the best metal that can be used for the mold, and it is expected that a cheap way to make such mold will be afforded by the Le Fer or Eustis method of electrolytically depositing 99.9 per cent pure iron in the form of pipes over a rotating mandrel, a good illustration of how one discovery helps another.

APPENDIX No. 5

STRESSES IN MOLDS IN MACHINES FOR CENTRIFUGAL CASTING

92 The subject of stresses in molds of machines for centrifugal casting is particularly important in large machines and in all machines using a hot mold, in the former because of the great numerical value of these stresses, and in the latter because of the low strength of materials at temperature of the order of 2000 deg. fahr.

93 These stresses depend primarily on the method of supporting the mold in the spinning bench. There are always present stresses:

a What might be called fluid-pressure stress, or the stress exerted by the liquid metal of the casting, and

b Flywheel stress, or the stress induced in the mold itself due to its rotation, and which would have been present even if it were spun empty. In addition to this, however, in a mold supported at intervals, there are present what might be called crankshaft stresses, stresses of bending essentially analogous to those induced in a crankshaft similarly supported. There are also gyroscopic stresses, the numerical value of which is, however, small as compared with the other three kinds.

94 The value of the crankshaft stresses is a matter of considerable importance, particularly as the deformations produced by these stresses are cumulative. For permanent molds in machines for centrifugal casting, materials have to be used which have a fairly low elastic limit, and this elastic limit is especially low at high temperature. Deformations produced in the course of spinning acquire, therefore, as a rule, the character of permanent sets, and as such affect the dynamic balance of the rotating parts, causing an unbalance which tends to increase the local stressing of the material, and this, in its turn, tends to increase the deformation. As the process is cumulative, it may, in the final end, lead to rapid destruction of the mold, and if not guarded against by proper caution, to accidents. It is therefore highly desirable, especially with hot molds, to avoid constructions in which crankshaft stresses may arise. The methods of calculation of these stresses are not given here, as they do not differ from those used by designers of steam turbines, with the exception of the fact that the bending strength of the material is very low as compared with the stress imposed.

FLUID-PRESSURE STRESSES

95 The stress produced by the molten metal in the mold of a machine for centrifugal casting is exactly of the type shown in Fig. 1 of the paper of Prof. Reid P. Stewart in the TRANSACTIONS of The American Society of Mechanical Engineers, vol. 34, 1912, p. 298.

96 The designer of a mold for a centrifugal casting machine is usually interested in selecting a safe thickness of wall for a given casting. As the size of the casting is given, the inside diameter of the mold is determined. Further-

more, as will be seen clearer later, the stress per square inch of the wall of the mold is also determined by the data of the problem. There is a certain leeway as to the permissible fiber stress in the material of the mold, if there are several materials to choose from, but once the material is selected and the temperature at which the mold is to be used is established, the fiber stress is also settled and all that remains is to determine the thickness of the wall of the mold.

97 For this purpose one of two formulas may be used, namely, the common formula

$$t = \frac{\frac{1}{2}Dp}{f} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [15]$$

or the Barlow formula

$$t = \frac{1}{2}\frac{D'p}{f} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

where t is the thickness of wall in inches, D the inside diameter in inches, D' the outside diameter in inches, p the internal gage pressure in lb. per sq. in., and f the permissible fiber stress in the wall in lb. per sq. in.

98 The "gage pressure" p of the Barlow formula is derived for centrifugal casting as follows. Assume that the casting has an external diameter D and an internal diameter d; the weight of a ring, 1 inch wide, is then $W = k(D^2 - d^2)$, where k is a constant (0.220 for steel; 0.204 for cast iron; 0.235 for brass; 0.252 for copper, etc.). But, in spinning, each pound of metal generates a centrifugal force determined by the various elements discussed in Appendix No. 3, primarily number of revolutions and radius of spinning, so that each pound of metal in the casting exerts on the mold a force equal to M lb., and hence on a ring of the mold, 1 inch wide, there is exerted a pressure MW lb.; hence, the "gage pressure" or pressure per sq. in. p is $\frac{MW}{\pi D}$ where D is the inside diameter of the mold in inches.

99 The difference between the common formula [15] and the Barlow formula [16] is that the first is expressed in terms of the inside diameter D and the second in terms of the outside diameter D'. Barlow's formula is preferable because it is on the side of safety. On the other hand, however, it cannot be used directly in this case because it begs the question. As shown by the above method of calculating p, it contains the value of the inside diameter D. Therefore if D', which is the outside diameter, were known, it would not be necessary to resort to either the Barlow or the common formula, as t could be obtained by simply subtracting the inside diameter from the outside and dividing the result by two. This does not prevent one from using the Barlow formula on condition, however, that it be expressed in a way slightly different from the usual, namely,

$$t = \frac{1}{2}\frac{p}{f}(D + 2t) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [17]$$

100 The meaning of either t or f in this formula should be very clearly understood; t means the thickness of the cylinder in which the highest permissible fiber stress is f that would withstand the pressure exerted by the centrifugal action of the molten metal on the mold under the assumption that the mold itself is not subject to centrifugal stresses of its own. In other words, t is that amount by which the thickness of the wall of the mold has to be increased in order to take care of the stress produced by the presence of the metal.

101 The meaning of t and f can be presented otherwise: t is the value of the thickness of the wall of the mold having a permissible fiber stress f which

the mold would have had if it had no mass of its own. Since, however, it has a mass and since, in usual practice, the weight of the mold considerably exceeds the weight of the casting, the stresses induced by the rotation of the mold itself have to be considered. These are what has been denoted by the term "flywheel stresses."

FLYWHEEL STRESSES

102 If the mold were running empty, that is, without any molten metal in it, it would act practically like an ideal flywheel, that is, a flywheel of considerable length having only a rim but without any arms.

Let w = weight of rim per inch of length in lb.

x = mean radius of rim in in.

N = revolutions per minute

then the centrifugal force acting on the entire rim which is here assumed to be one inch long is

$$P = 0.000341 \frac{wxN^2}{12} \quad \text{[18]}$$

103 This is the familiar formula expressed, however, in inches of radius instead of feet.

104 The same formula may be expressed, as follows:

$$P = 0.000014\ wD''N^2 \quad \text{[19]}$$

where P is the total centrifugal force on the rim and D'' is the diameter in inches.

105 The question now is which diameter to take. In flywheels the thickness of the wall T is very small as compared with the diameter, and therefore the mean diameter or mean radius can be taken for purposes of calculation without introducing a greater degree of inexactness than is carried by the formula itself. In molds for machines for centrifugal casting the situation, however, is very different in this respect, the thickness of the wall being very appreciable as compared with the diameter. Therefore it would not be safe to take the mean diameter, and following the practice of the Barlow formula, the outside diameter should be here taken. On the other hand, however, this would introduce the same difficulty as to the expression of T as has been encountered in considering the fluid pressure stresses. Therefore P may be here expressed as follows:

$$P = KwN^2(D + 2T) \quad \text{[20]}$$

where T is the thickness of the rim of the flywheel, as yet irrespective of the stresses induced therein, and $K = 0.000014$.

106 This is the total stress on a section of the mold one inch long treated as a flywheel, that is, running empty. In accordance with the usual processes, it is necessary to divide this by 6.28 to obtain the strain on the cross-section of the mold, in this case equal to T sq. in., and this gives $0.0000023\ N^2w(D + 2T)$, which could be written as $Sw(D + 2T)$ where $S = 0.0000023N^2$.

107 From this it would be possible to calculate the value of T, provided the fiber stress f is either known or assumed, since $Tf = Sw(D + 2T)$.

108 At a glance the total thickness of the mold could be obtained by adding T, as derived above, to t as derived in the preceding discussion of the fluid pressure. Actually, however, the value thus obtained would be smaller than the true value of stresses, because the part of the wall equal to t has centrifugal stresses of its own induced in it. Because of this, it is simpler to use a different method based on the following.

109 In the discussion of fluid pressure it was found that the total pressure produced by the fluid metal on a ring cut out of the mold one inch long is equal to WM pounds. This entire strain has to be taken up by the mold in the same way as centrifugal rim stresses are in a flywheel, which means that the strain induced is $\frac{WM}{6.28}$ which may be written as sWM. Hence the total strain

$$P' = T'f = Sw(D + 2T') + sWM \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]$$

where T' is the thickness of the mold sufficient to take up both the fluid pressure and the flywheel stresses, and f, as before, the highest permissible fiber stress in pounds per square inch. From this equations for either T' or f are derived.

$$T' = \frac{SwD + sWM}{f - 2Sw} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [22]$$

$$f = \frac{Sw(D + 2T') + sWM}{T'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

110 It should be noted in this connection that T' errs here on the side of safety, the entire material of both the mold and the casting being assumed as concentrated on the outside diameter of the mold rather than being distributed between the outside diameter of the mold and the inside diameter of the casting. This may introduce an error as high as 20 per cent. Since, however, this error is on the side of safety, it is not objectionable.

THERMAL-EXPANSION STRESSES

111 As the molten metal is pouring into the mold, the inner wall of the mold is rapidly heated up, and it takes a certain amount of time for the heat to spread throughout the metal and for the temperature of the various parts of the mold to become equalized. This time depends on the heat conductivity of the material of the mold, and in very thick molds may be sufficiently appreciable to induce thermal stresses in the mold, due to more rapid expansion of the inner wall of the mold than of the outer wall of the mold. While it is possible to compute these stresses by using the process developed some fifty years ago by Lord Kelvin for calculating the flow of heat in cylindrical bodies, the experimental data are not sufficient to obtain results that would be valuable in practice and these kinds of strains have to be simply assumed and taken care of in selecting the value for the highest permissible fiber stress, f. This is also one of the reasons why a process for estimating T' has been selected in such a way as to admit of a considerable error on the side of safety.

SPLIT MOLDS

112 If split molds are to be used it is obvious that the entire stress P' is taken up not by the wall of the mold, but by the elements holding the mold together such as bolts, or lugs and dovetails, if the construction shown in Fig. 4 is used. It is obvious that in such a case the thread of the bolts or the side lugs are in shear, the unit value of the shear being $\frac{P'}{h}$, where h is the number of the bolts or lugs that hold the mold together.

DISCUSSION

James B. Ladd. The writer has read the paper on centrifugal casting with unusual interest because he has devoted considerable time to the subject, and he feels sure that it will be appreciated by all who are interested in this line of engineering.

The writer believes that most readers of this paper, engineers as well as others, will get the impression that successful centrifugal casting, that is commercial centrifugal casting, can easily be accomplished, and he therefore regrets that the author does not emphasize the fact that it is an exceedingly difficult process to perfect, as is evidenced by the fact that it has been known for more than a century yet not developed on a commercial scale until quite recently, although it has received much attention in this country alone, as is shown by the 135 United States patents listed in Appendix No. 2.

This refers, of course, to the centrifugal casting of ferrous pipes and tubes to which the greater part of the paper is devoted, as the centrifugal process is appreciably less difficult for hollow castings of relatively short lengths and for non-ferrous metals.

It is stated in Par. 42 that "notwithstanding the great simplicity of the machinery and methods of centrifugal casting, it should be clearly remembered that even the slightest imperfections in the design or operation will immediately show up in the casting," but this comes so late in the paper that it may not correct the impression received by readers from several preceding paragraphs, especially Par. 3, which states in part "there is no reason why any good engineer could not design and operate a successful casting machine," which of course means a centrifugal casting machine.

While some of the difficulties to be encountered in developing a centrifugal process commercially are stated in this paper, the writer feels that such difficulties are not sufficiently emphasized and that it should be clearly stated that no engineer should undertake to develop a centrifugal casting process without knowing in advance that he should not expect commercial results without the expenditure of much time and much money. Further, that he should not only become familiar with the patent situation, as

the author states, but that he should also carefully watch the patent situation, because, even if he should get satisfactory results, he is likely to find that he has been antedated in essential details, not only by issued patents but also by unissued patents.

That difficulties are to be encountered in developing a centrifugal process is well illustrated by the history of the making of cast-iron pipe, which was made by the centrifugal process as early as about 1850, and yet the de Lavaud process of making cast-iron pipe was not perfected commercially until a few years ago, and it stands alone today as the only centrifugal process that makes cast-iron pipe commercially.

The term " commercial process " is used as defining a process which produced a product at a cost not greater than the cost of the same product of equal quality produced by other processes. In this connection it is interesting to note that a commercially perfected centrifugal process not only produces a product competitive in cost and quality with like products by other processes, but that usually the centrifugal product is of superior quality. Thus, in the case of the de Lavaud process cited by the author, the strength of the iron in the pipe is greatly increased over the strength of the same iron in pipes cast in sand by the usual foundry method. The author gives this increase in strength conservatively as about 15 per cent, but the writer believes that it is more nearly 40 or 50 per cent.

From Par. 20 the writer understands that the " Cammen process " is one for making steel pipe and tubes in highly heated centrifugal molds. If, as related in Par. 21, this process involves the removal of the castings and the molds from the centrifugal machine at " a temperature well above a white heat," the writer fears that it is likely to be disappointing, as it is difficult to keep hot molds in perfect running condition without removing them from the machine, and the difficulties will be increased if the molds must be handled when hot.

The statement in Par. 7 that a permanent mold is the only practical way of producing pipe on a tonnage basis indicates that the author, like others, has experienced the difficulties inherent to molds lined with sand, which difficulties include the handling of the molds to and from the centrifugal machine and the necessity of using a large number of molds for quantity production, and yet the " Cammen process " includes handling *white-hot molds.*

The description of the method of operation of the Cammen process given in Par. 21 starts with the castings within the molds and omits entirely to describe how the metal is introduced into the mold, but Par. 67 seems to indicate that the metal is poured in at one end of the mold and is spread the entire length of the mold by centrifugal force.

If the metal is poured in at one end of the mold, it is not surprising that he considers it advisable to heat the mold highly. The writer believes that satisfactory results are to be obtained with less highly heated molds by using a moving runner and distributing the metal throughout the mold by the de Lavaud process.

W. M. Corse. The contribution of such a complete paper marks a distinct step in advance in the recording of the art of this very important commercial process. The author has not only given the history of this process but has described many of its salient features in detail. It seems to the writer that we should be interested in centrifugal casting because it is so practical commercially. The speed of production far exceeds that of the ordinary sand-molding method and the quality of the casting is very superior to any casting that it is possible to produce by the sand-molding method. As the author says, there have been persistent efforts to keep out of the literature information on this process. With all the facts given it becomes evident that obstructive tactics on the part of any manufacturer of centrifugal castings is not only a commercial error but also an error in judging the present state of the art. The author has done the manufacturing industry a service by accumulating so much information regarding the early history of this process as described in Par. 3, namely: "It would appear, therefore, that with the exception of minor details and one or two features in special casting processes there is no reason why any good engineer could not design and operate a successful casting machine with a non-heated or non-cooled mold without running into legal complications." This statement should set straight the minds of many people who have been interested in this subject.

The Author appreciates fully the weight of the remarks of Mr. Ladd. It is no doubt true that a great deal of knowledge is necessary for successfully designing centrifugal casting machinery and when the writer stated in Par. 3 that there is no reason why

any good engineer could not design and operate a successful casting machine he had reference only to the patent situation and not to its engineering features.

At the same time, it is well not to exaggerate the difficulties of designing such machines. Thus, excellent machines have been designed and operated by the engineers of the Brooklyn Navy Yard with only a very moderate amount of preliminary experimental work.

As regards Mr. Ladd's remarks as to the great difficulty of handling white hot centrifugal molds, the writer wishes to state that he did not find any trouble in this connection, which he ascribes to proper design of the molds and especially to the use of proper materials. Had the molds been made of cast iron or ordinary steel they would, of course, fly to pieces at the first spinning, but they stand up very well when made of proper alloys. Also, the molds must be properly supported in handling, a problem which has been solved without much trouble. As regards Mr. Ladd's fears that the Cammen process " is likely to be disappointing " because of the difficulty of keeping the hot molds in perfect running condition, etc., it may now be stated that actual experience has shown that such fears are entirely without foundation.

As far as the writer is aware the De Lavaud process has not been used for steel, and, in fact, it is doubtful if it could be so used. Neither does it seem applicable to such metals as monel, nickel and chrome-nickel alloys, where the pouring must be extremely rapid and effected at temperatures of the order of 3000 deg. fahr. No trouble has been experienced with such castings in properly preheated molds.

No. 1848

HEAT LOSSES FROM BARE AND COVERED WROUGHT-IRON PIPE AT TEMPERATURES UP TO 800 DEG. FAHR.

By R. H. Heilman, Pittsburgh, Pa.
Junior Member

High-temperature superheated steam running up to 800 deg. fahr. and high-temperature chemical processes are being more and more widely used, and accordingly the question of heat losses from pipes under such temperature conditions is one of importance to the engineering profession.

This paper presents the findings of an experimental investigation conducted in the Mellon Institute of Industrial Research of the University of Pittsburgh. The losses from bare wrought-iron pipes have been measured for temperatures up to and including 800 deg. fahr. They have been studied carefully for pipes of various diameters, and empirical formulas are presented whereby the loss from pipes of any diameter may be readily calculated.

THE purpose of this paper is to report some of the data obtained recently on bare pipes operating at temperatures up to 800 deg. fahr., and to present curves and formulas which will enable the engineer to solve readily the problems usually encountered in the calculation of heat losses from bare and covered pipes.

BARE-PIPE HEAT LOSSES

2 Manufacturers of pipe coverings often are required to guarantee that the application of a specified heat-insulating covering will effect a certain percentage saving of the heat which would be lost entirely from a bare pipe. Since the bare-pipe loss is the 100 per cent value against which the losses from the covered pipe must be compared, it is essential that the loss from the bare pipe shall be known accurately.

3 Many investigators have studied the heat losses from bare pipes. Perhaps the most noteworthy of these experimentalists was the French physicist Péclet. Paulding, in his book on Steam in

Presented at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers. Awarded A. S. M. E. Junior Prize for the best paper during the year 1922.

Covered and Bare Pipes, has worked out the theory of heat losses from bare pipes in the light of the researches of Péclet. However, the findings of later investigators do not support the results given by Paulding. Owing to the fact that Péclet's experiments were conducted at very low temperatures, while subsequent investigators confined themselves mostly to one pipe size only, the Mellon Institute deemed it advisable to carry on the research to higher temperatures and to pipes of various sizes. By testing pipes of various sizes under the same condition, it was thought that more reliable data could be secured than by comparing the results of other investigators on pipes of various sizes under different conditions. Accordingly, tests were made on 1-in., 3-in., and 10-in. pipe.

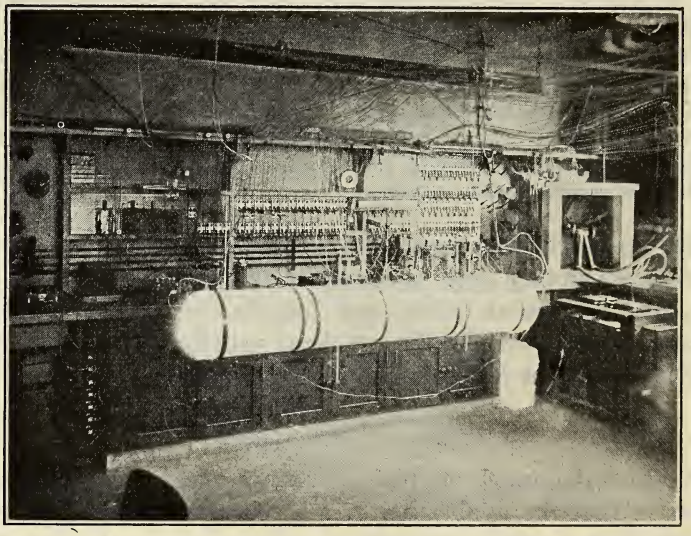

Fig. 1 Laboratory and Testing Apparatus

4 The method of testing was practically the same as that described by G. D. Bagley in his paper on Conversion of Heat Losses from Pipes and Boilers, presented before the Society in 1918, except that the pipe-covering tester had been improved. A Leeds & Northrup type K potentiometer was substituted for the milli-voltmeter, and a General Electric saturated-core-type voltage regulator and smaller thermocouples were added. Fig. 1 shows a view of the laboratory and the testing apparatus.

5 When conducting tests on bare pipes it is very desirable that

the room temperature remain constant throughout the work, for the rate of heat loss is dependent upon the absolute temperature as well as upon the temperature difference. The 1-in. pipe was selected for test at the higher temperatures, as the relatively small amount of heat loss from a 1-in. pipe could not greatly affect the room temperature.

6 This pipe was run up to a temperature of 800 deg. fahr. The average room temperature throughout this test was 81 deg. fahr. and the temperature did not vary more than 1.8 deg. fahr. during its progress.

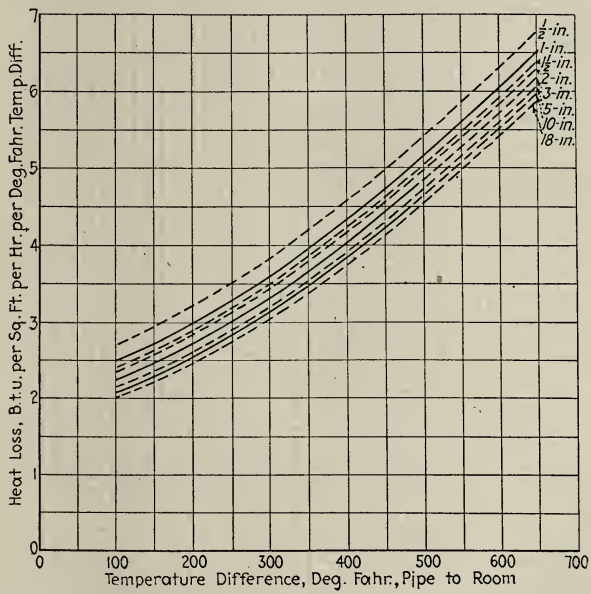

Fig. 2 Bare-Pipe Loss Curves

7 When the emissivity coefficient is plotted against the heat loss, it is found that the curve obtained is not a straight line, but falls off at the higher temperatures. This means that the heat loss from a bare pipe does not increase as rapidly at the higher temperatures as would be anticipated. A possible explanation for this condition is that the convection loss at the higher temperatures does not increase as rapidly as at the lower temperatures.

8 The first test on the 1-in. pipe was checked by a second test on another 1-in. pipe and both tests corresponded exactly at the higher temperatures.

Gage Press	Hot Water			10 Lbs.			80 Lbs.			120 Lbs.			160 Lbs.			200 Lbs.			200 Lbs. and 100° F. Superheat.			275 Lbs. and 250° F. Superheat.		
Temp.	180° F			239.4° F.			324.0° F.			350.0° F.			370.7° F.			387.9° F.			487.9° F			664.3° F.		
Pipe Size	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.	Dollars Loss	Pounds Coal	B.T.U. per Lineal Ft. per °F. Diff. per Hr.
½"	1.32	526	.605	2.24	897	.670	3.92	1566	.779	4.51	1805	.815	5.25	2010	.846	5.53	2210	.875	8.60	3440	1.040	16.15	6460	1.375
1"	1.91	763	.878	3.26	1305	.973	5.72	2290	1.138	6.50	2601	1.178	7.27	2910	1.242	8.07	3230	1.280	12.56	5026	1.520	23.84	9540	2.030
1½"	2.65	1060	1.220	4.54	1818	1.357	7.98	3190	1.586	9.28	3710	1.676	10.42	4165	1.751	11.36	4550	1.804	17.94	7175	2.170	33.76	13500	2.870
2"	3.24	1297	1.491	5.36	2142	1.600	9.78	3910	1.943	11.37	4549	2.052	12.75	5100	2.145	14.00	5600	2.220	22.10	8825	2.670	42.20	16890	3.590
2½"	3.86	1545	1.778	6.65	2660	1.984	11.66	4660	2.320	13.66	5460	2.464	15.25	6100	2.564	16.70	6690	2.650	26.64	10650	3.220	50.50	20200	4.300
3"	4.56	1824	2.100	8.24	3292	2.460	13.88	5550	2.760	16.14	6450	2.910	18.04	7210	3.030	19.84	7945	3.150	31.60	12640	3.820	58.25	24300	5.170
3½"	5.18	2070	2.380	8.89	3554	2.655	15.82	6325	3.145	18.31	7322	3.305	20.50	8200	3.450	22.60	9040	3.580	35.90	14360	4.345	69.00	27600	5.860
4"	5.78	2305	2.650	9.87	3950	2.950	17.70	7075	3.520	20.50	8200	3.700	22.85	9145	3.842	25.15	10060	3.981	40.10	16040	4.850	72.40	30920	6.580
4½"	6.35	2540	2.920	10.94	4370	3.260	19.48	7790	3.878	22.60	9025	4.075	25.30	10120	4.250	27.75	11100	4.400	44.05	17620	5.326	85.60	34240	7.286
5"	6.95	2780	3.200	11.97	4790	3.575	21.25	8500	4.232	24.62	9850	4.450	27.60	11050	4.650	30.40	12140	4.805	49.35	19740	5.960	94.50	37800	8.050
6"	8.20	3280	3.775	14.21	5680	4.240	25.30	10110	5.024	29.30	11720	5.295	32.80	13120	5.522	36.00	14420	5.715	57.50	23000	6.955	112.50	45000	9.580
7"	9.40	3760	4.325	16.18	6470	4.826	29.10	11640	5.782	33.70	13480	6.090	37.60	15040	6.324	41.60	16650	6.560	66.25	26500	8.010	129.00	51550	10.97
8"	11.00	4398	5.050	18.25	7300	5.450	32.60	13030	6.455	37.65	15050	6.840	42.40	16940	7.125	46.05	18420	7.300	74.75	29900	9.050	146.20	58500	12.44
9"	11.62	4650	5.350	20.35	8130	6.070	36.25	14500	7.210	42.10	16840	7.600	47.22	18900	7.950	51.98	20790	8.230	83.50	33400	10.10	162.60	65050	13.84
10"	12.68	5065	5.925	22.05	8820	6.584	40.20	16100	8.010	46.70	18690	8.440	52.30	20910	8.805	57.75	23100	9.150	97.50	39000	11.18	180.00	72000	15.30
12"	15.00	6000	6.995	26.44	10580	7.890	47.40	18950	9.425	55.40	22120	10.00	61.76	24700	10.40	70.00	28010	11.14	109.30	43700	13.22	213.50	85500	18.18
14"	16.60	6635	7.625	28.90	11560	8.620	52.00	20800	10.34	60.50	24200	10.92	67.50	27000	11.36	74.10	29650	11.76	119.40	47740	14.43	233.00	93300	19.85
16"	18.82	7525	8.650	32.80	1312.0	9.790	58.76	23500	11.70	68.40	27320	12.34	76.10	30410	12.80	84.25	33700	13.35	135.00	54000	16.34	265.00	106000	22.55
18"	21.00	8400	9.650	36.10	14460	10.80	65.40	26150	13.00	76.50	30570	13.80	85.50	342.00	14.38	93.75	37500	14.87	151.20	60500	18.30	297.50	119000	25.30

TABLE 1 LOSSES FROM HORIZONTAL BARE-IRON STEAM PIPES

From 100 Lineal Feet of Pipe per Month of 30 Days with Steam in Pipes 24 Hr. per Day. Coal at $4.00 per Ton of 2000 Lb.

In this table coal has been figured at $4.00 per ton of 2000 lb., 13,000 B.t.u. per lb. of coal; labor, boiler-room expense, etc., taken at $1.00 per ton, making total value of coal fired at $5.00 per ton. Boiler efficiency taken at 70 per cent; air temp. 70 deg. fahr. Experimental data obtained at the Mellon Institute.

9 The location of the curves for the 3-in. and the 10-in. pipes was obtained by experiment at the lower temperatures, as indicated by the solid lines in Fig. 2. The values for the higher temperatures are the result of extending the curves parallel to the curve obtained for the 1-in. pipe. This procedure was necessary because of the fact that the larger pipes could not be raised to the higher

TABLE 2 HEAT LOSSES FROM HORIZONTAL BARE-IRON PIPES

*Heat Losses from Horizontal Bare Iron Pipes.													
Temperature Difference Between Pipe Surface & Air Surrounding Pipe													
Pipe Size Inches	50°	100°	150°	200°	250°	300°	350°	400°	450°	500°	550°	600°	650°
½	2.48	2.70	2.95	3.21	3.51	3.84	4.20	4.57	4.97	5.41	5.86	6.31	6.78
¾	2.36	2.57	2.80	3.07	3.36	3.68	4.04	4.42	4.81	5.25	5.70	6.15	6.62
1	2.29	2.50	2.73	3.00	3.29	3.60	3.95	4.34	4.73	5.16	5.60	6.05	6.51
1¼	2.24	2.45	2.68	2.94	3.23	3.54	3.89	4.28	4.67	5.09	5.54	5.98	6.44
1½	2.20	2.41	2.64	2.90	3.19	3.50	3.85	4.23	4.61	5.03	5.48	5.92	6.39
2	2.15	2.35	2.58	2.85	3.14	3.45	3.79	4.16	4.56	4.98	5.41	5.84	6.30
2½	2.09	2.29	2.52	2.78	3.06	3.37	3.71	4.09	4.49	4.91	5.32	5.78	6.24
3	2.05	2.25	2.47	2.73	3.00	3.31	3.65	4.03	4.43	4.85	5.26	5.71	6.19
3½	2.02	2.22	2.43	2.69	2.97	3.27	3.61	3.98	4.38	4.80	5.22	5.67	6.15
4	2.00	2.19	2.40	2.66	2.94	3.24	3.58	3.95	4.35	4.77	5.19	5.64	6.11
4½	1.98	2.17	2.38	2.63	2.92	3.22	3.56	3.92	4.32	4.74	5.17	5.62	6.07
5	1.97	2.15	2.36	2.61	2.90	3.20	3.54	3.90	4.30	4.71	5.15	5.60	6.05
6	1.95	2.13	2.34	2.59	2.88	3.18	3.52	3.89	4.28	4.69	5.12	5.57	6.02
7	1.93	2.11	2.32	2.57	2.86	3.16	3.50	3.87	4.26	4.67	5.10	5.55	6.00
8	1.91	2.09	2.31	2.56	2.84	3.14	3.49	3.86	4.25	4.66	5.09	5.53	5.99
9	1.90	2.08	2.30	2.55	2.83	3.13	3.48	3.85	4.24	4.65	5.08	5.51	5.98
10	1.89	2.07	2.29	2.54	2.82	3.12	3.47	3.84	4.23	4.64	5.07	5.50	5.97
12	1.87	2.05	2.27	2.52	2.80	3.10	3.45	3.81	4.20	4.61	5.05	5.48	5.94
14	1.86	2.04	2.25	2.50	2.78	3.08	3.43	3.79	4.18	4.59	5.03	5.46	5.92
16	1.85	2.03	2.23	2.48	2.76	3.06	3.41	3.77	4.16	4.57	4.99	5.44	5.90
18	1.84	2.02	2.22	2.47	2.75	3.05	3.39	3.75	4.14	4.55	4.98	5.42	5.88

* B.T.U. per Sq. Ft. Pipe Surface, per Hr. per Deg. Fahr. Temp. Diff. Air Temp. 70 Deg. Fahr.

temperatures without raising considerably the temperature of the room. Tests are now in progress to ascertain the loss from vertical iron pipes, and the results will be reported later.

10 In Table 1 the loss in dollars and cents and in pounds of coal per 100 lineal feet of horizontal bare-iron pipe is tabulated for temperatures up to 664 deg. fahr. The loss varies from $1.32 for 100 lineal feet of ½-in. pipe at 180 deg. fahr. to $297.50 for 100 lineal feet of 18-in. pipe at 664 deg. fahr. A ten-degree fahr. temperature

drop has been assumed from the steam to the outer surface of the pipe for superheated steam.

THEORY OF HEAT LOSS FROM INSULATED PIPES

11 In order to calculate the loss of heat from an insulated pipe or boiler, it is necessary to know the total temperature drop from the pipe to the surrounding air; and to enable one to make accurate calculations it is required that the component temperature drops be known.

12 The total temperature drop from the steam inside a pipe to the outer air can be considered as made up of four components, as follows:

(*a*) Drop from steam to the outer surface of the pipe

(*b*) Drop from outside surface of the pipe to inside surface of the insulation

(*c*) Drop from the inside surface of the insulation to the outside surface of the insulation

(*d*) Drop from outside surface of the insulation to the surrounding air.

13 The temperature drop from the steam to the outer surface of the pipe depends upon the resistance to heat flow offered by the film at the inner surface and the resistance offered by the iron wall of the pipe.

14 This combined resistance is very low, owing to the high conductivity of the iron and the relatively high conductivity of the water film. Consequently the temperature drop is very low. No attempt has been made to measure the above temperature drop in this investigation, as it was considered to be so small as to be negligible. This drop has been measured for saturated steam by L. B. McMillan [1] and found to be a fraction of a degree. However, a test conducted by Eberle [2] for superheated steam shows a drop as high as 10 deg. fahr. This drop should be taken account of when making calculations for superheated steam.

15 The temperature drop from the outside surface of pipe to the inner surface of the insulation depends upon the resistance to heat flow offered by the air space between the surface of the pipe and the inner surface of the insulation. The heat flow which takes place in this case is due to radiation, conduction, and convection. Since the radiation increases as the fourth power of the absolute

[1] Trans. Am. Soc. M. E., vol. 37, p. 928.

[2] Mitt. über Forschungs-Arbeiten auf dem Geb. des Ing., heft 78.

temperature difference, it is to be expected that the temperature drop would tend to decrease at the higher temperatures.

16 The temperature drop from the outer surface of the pipe to the inner surface of the covering for 1-in., 3-in., and 10-in. pipe is shown in Fig. 3. These curves show that the temperature drop increases as the pipe diameter decreases. A test was also made on a 3-in. pipe with an air space of 1.2-in. between the surface of the pipe and the insulation. By comparing this curve with the curve for an air space of 0.1 in., it is observed that the temperature drop

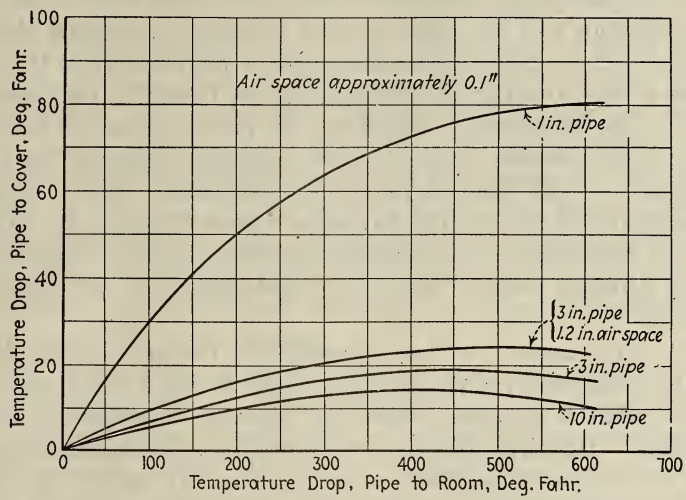

Fig. 3 Temperature Drop from Outer Surface of Pipe to Inner Surface of Covering

for a 1.2-in. air space is only a few degrees more than for an air space of 0.1 in. This is probably due to the fact that for air spaces much greater than 0.2 in. convection currents are increased, thus causing an increase in heat loss.

17 Since the flow of the heat is directly proportional to the cross-sectional area, and inversely proportional to the length of the path, it is obvious that the presence of a 1.2-in. air space between the surface of the 3-in. pipe and the inner surface of the insulation will cause an increase in the total amount of heat lost from the surface of the pipe, provided, of course, the air space is not as good an insulator as the covering itself. In this case the insulating value of commercial coverings is many times greater than the insulating value of the air space.

18 The results of this test show that an air space of over 0.25 in. is of little use as an insulator on flat surfaces at high temperatures, and that this air space is of little value as a protection to the covering from the effects of the high temperatures. This test also demonstrates that coverings should be kept as close as possible to cylindrical surfaces, because the insertion of an air space of approximately 0.1 in. between the pipe and insulation actually increases the overall loss. An examination of Fig. 3 shows that the temperature drop for a 0.1-in. air space is approximately equal to that for 0.1 in. of commercial insulations, so that this temperature drop can be neglected in calculations and the pipe covering considered as fitting close to the pipe with the pipe temperature and the temperature at the outer surface of the covering as the temperatures bounding the covering.

19 The temperature drop from the inside surface of the insulation to the outside surface of the insulation depends upon the resistance to heat flow offered by the insulation itself. Heat is transmitted through the insulation by means of radiation, conduction and convection. The relative amount of each of these three factors depends entirely upon the construction of the insulating material.

20 The amount of heat transmitted through the insulation and the temperature drop from the inner to the outer surface determine what is generally called the absolute conductivity of the insulation. However, this does not give the true absolute conductivity of the insulation, but gives what may be called the mean absolute conductivity. The true absolute conductivity for an insulating material, say, 1 in. in thickness can be represented by a curve. The absolute conductivity for a given material increases as the temperature increases, and therefore the absolute-conductivity curve depends upon the thickness of the covering and also upon the curvature of the covering.

21 The drop in temperature, or the temperature-gradient curve through the insulation, then depends upon the thickness of the covering and the curvature. In a cylindrical covering the resistance to heat flow diminishes as the outer surface is approached, the temperature drop becomes less, and the gradient curve is bowed downward if the curvature alone is taken into consideration. However, the absolute conductivity decreases as the outer surface is neared, with a consequent bowing up of the gradient curve, and the two tend to counteract each other, so that the temperature-gradient curve may be bowed either up or down or be a straight line, depend-

ing upon the curvature of the cylinder. The temperature-gradient curve for a flat surface should bow up.

22 It is highly desirable that tests should be conducted on commercial steam-pipe coverings of different thicknesses and at different temperatures, in order to obtain mean absolute-conductivity curves for the different thicknesses.

23 The temperature drop from the outer surface of the insulation to the surrounding air depends upon the amount of heat emitted by radiation and air contact. This in turn is dependent upon the nature of the surface of the body, the shape of the body, the excess of its temperature over that of objects to which radiation takes place, and the absolute value of the temperature of these bodies. Commercial steam-pipe coverings are invariably covered with a canvas jacket. From the above-mentioned facts it is obvious that the loss from a canvas surface at a given temperature is independent of what is under the canvas, so that, if the canvas-loss law can be ascertained, this law may be applied to the loss from steam-pipe coverings and thus the temperature of the outer surface of the insulation can be determined. In making calculations of heat loss through an insulation, it is absolutely necessary to know the temperatures at the inner and outer surfaces.

24 Péclet made a careful study of the heat emissivity from various surfaces, canvas surfaces included. As mentioned, however, his experiments were conducted at relatively low temperatures. McMillan made a study of the heat emissivity from a canvas surface in his study of commercial steam-pipe coverings, but he confined his experiments to one pipe size only. Nevertheless, McMillan's results in the form of a curve present a readier means of calculating the losses from steam-pipe coverings than do Péclet's, whose observations, while taking all the variables into consideration, are in too complicated a form to provide a ready means of calculation.

25 Since McMillan's canvas-surface-loss curve was obtained from experiments on one pipe size only, this curve can be used in making calculations on coverings of a diameter approximately the diameter of the coverings tested. In order to be able to calculate the loss of heat from pipe coverings of any diameter, it has been necessary to obtain the canvas-surface-loss curves for various diameters. Accordingly, coverings were tested on the 1-in., 3-in., and 10-in. pipes used in determining the bare-pipe losses. The average outer diameters of the coverings used were 3.1 in., 9.5 in. and 17.2 in. The results of these tests are shown in Fig. 4.

26 In order to simplify the calculations necessary to determine the loss of heat through coverings of various diameters, the equations of the three curves shown in Fig. 4 have been derived. In these equations—

T_d = temperature difference between canvas surface and room, deg. fahr.

h = total B.t.u. loss per hour per square foot of canvas surface

D = outer surface diameter, inches.

$$\text{3.1 in. diameter: } T_d = \frac{272.5\,h}{h + 456.5} \quad \ldots\ldots \quad [1]$$

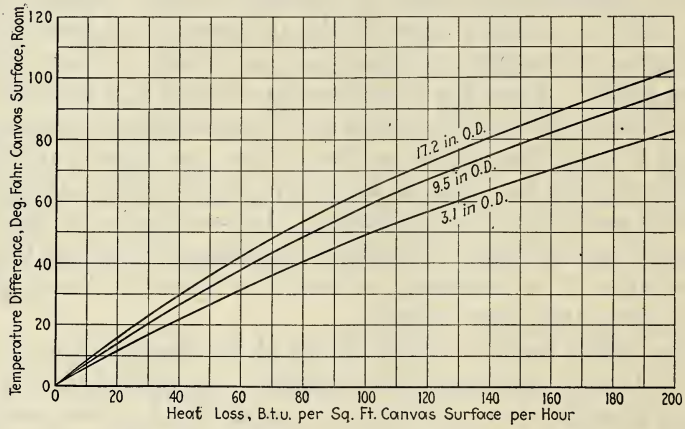

FIG 4 CANVAS-SURFACE-LOSS CURVES

$$\text{9.5 in. diameter: } T_d = \frac{272.5\,h}{h + 368} \quad \ldots\ldots \quad [2]$$

$$\text{17.2 in. diameter: } T_d = \frac{272.5\,h}{h + 328.4} \quad \ldots\ldots \quad [3]$$

These equations can be combined to give the general equation:

$$T_d = \frac{272.5\,h}{h + \dfrac{564}{D^{0.19}}} \quad \ldots\ldots\ldots\ldots \quad [4]$$

which is approximately accurate for diameters up to 2 ft.

27 It is believed that these curves are fairly accurate, inasmuch as they were obtained from the results of numerous tests on different materials.

28 Thermocouples were used in determining the canvas temperatures. During this investigation, it was found that the couple,

when just inserted under the canvas, would invariably read low. This difficulty was overcome by inserting it under the canvas for a distance of several inches, this distance depending upon the size of the couple and the temperature of the covering. From a theoretical consideration of the question, it can be shown that the minimum distance to which the thermocouple can be placed under the canvas is reached when the temperature of the thermocouple wires, a short distance from the junction, is the same as the temperature at the junction. When this condition is reached, there is no flow of heat from the junction to the wires and consequently no lowering of the junction temperature.

SAMPLE CALCULATIONS

29 A covering 2 in. thick, assumed as having a mean absolute-conductivity coefficient of 0.56, is placed on a 4½-in. outside-diameter pipe maintained at a temperature of 400 deg. fahr. The temperature of the surrounding air is 70 deg. fahr. Determine the heat flow in B.t.u. per hour per sq. ft. of pipe surface.

30 The heat flow through a cylinder is given by the equation:

$$H = \frac{K(T_1 - T_2)}{r_1 \log_e \frac{r_2}{r_1}} \quad \text{. . . . . [5]}$$

where T_2 is the temperature at the outer surface of the covering. To obtain T_2, knowing only T_1, the pipe temperature and T_3, the room temperature, it is necessary to change the form of the equation so as to include T_d. The equation for T_d, as developed from experimental results, is

$$T_d = \frac{272.5\,h}{h + \frac{564}{D^{0.19}}} \quad \text{. . . . . [4]}$$

whence

$$h = \frac{K(T_1 - T_3 - T_d)}{r_2 \log_e \frac{r_2}{r^1}} \quad \text{. . . . . [6]}$$

in which

h = B.t.u. loss per hour per sq. ft. of canvas surface
K = mean absolute conductivity of insulation
r_1 = radius of inner surface of insulation, inches
r_2 = radius of outer surface of insulation, in inches
T_d = temperature difference between outer surface of insulation and room, deg. fahr.

31 Substituting in [6], there results

$$h = 0.56\left[\frac{400 - 70 - \dfrac{272.5\,h}{h + \dfrac{564}{8.5^{0.19}}}}{4.25 \log_e \dfrac{4.25}{2.25}}\right]$$

$$= 0.207\left(330 - \frac{272.5\,h}{h + 375}\right)$$

$$= \frac{-363.1 \pm \sqrt{363.1^2 + 4 \times 25{,}600}}{2} = 60.4$$

$$H = 60.4 \times \frac{4.25}{2.25} = 114 \text{ B.t.u. loss per hr. per sq. ft. of pipe surface.}$$

33 The temperature at the outer surface of the covering can be obtained by substituting the value of h in the equation $T_d = \dfrac{272.5\,h}{h + 375}$, reducing to $T_d = 37.6$. Therefore the temperature of the outer surface of the covering is

$$37.6 + 70 = 107.6 \text{ deg. fahr.}$$

DISCUSSION

B. N. Broido. The author states that the temperature drop between the steam and the wall of the pipe is very low, that it has been found to be, for saturated steam, a fraction of a degree, and with superheated steam, in accordance with tests by Eberle in Munich, only 10 deg. This is misleading, as it might give the reader the impression that the temperature between the superheated steam and the wall of a bare pipe is only 10 deg. As a matter of fact, Table 3, which shows the result of the tests with superheated steam in covered and bare pipes, shows a temperature difference at a velocity of the steam of about 30 ft. per sec., and, at a pressure of 98 lb., of as high as 110 deg.

Of particular interest is column 6 of the table, which shows the heat transfer from the steam to the metallic wall per sq. ft. and

1 deg. temperature difference. The lowest is 15.4 B.t.u. at 30 ft. velocity; the highest, 36 B.t.u. at a velocity of about 98.5 ft.; while the same heat transfer for saturated steam is over 400 B.t.u. This shows the difference in the heat transfer from the steam to the pipe between superheated and saturated steam.

It is true that this difference is of less importance for a covered pipe, and the better the insulating quality of the covering, the less important is this difference. There is, however, no covering which is a perfect insulator, and there are a great many power plants where the piping is very poorly covered, so that the property of superheated steam of not readily giving up its heat is advantageously utilized, and in many cases, especially with long pipe lines, the saving due to elimination of radiation losses in itself warrants installation of a superheater.

With the exception of the Munich tests, no others have been made, to the knowledge of the writer, to show the difference in radiation losses from pipe lines, between saturated and superheated steam, due to the fact that it is very difficult to measure the radiation losses of saturated steam. With superheated steam, the radiation losses are expressed in the temperature drop of steam, while with saturated steam, no temperature drop occurs, and a part of the steam is condensed, and there are no means to completely separate the water from the steam. Practice has shown, however, that there is considerable reduction in the radiation losses in a pipe line carrying superheated steam, as compared with one with saturated steam.

In the table of losses from bare iron steam pipes, the author makes the same error as many of the pipe-covering manufacturers in giving the pounds of pressure of the steam and the superheat in degrees fahrenheit, corresponding to the total temperature, which gives the impression that the tests were made with superheated steam actually flowing through the pipe, while as a matter of fact these tests are made with electrically heated pipe. The radiation losses as well as the temperature of the wall, if superheated steam of the given temperature and superheat were flowing through the pipe, would be considerably less.

Another point which should not be neglected, but which is not mentioned, however, is the influence of the velocity and moisture contents of the air. The author properly states that when conducting tests on pipes, it is very desirable that the room tempera-

ture remain constant. In order to have a constant temperature, the air in the room is kept as still as possible, which to a certain extent defeats the purpose of the tests, as in practice, air is always moving more or less, and the results are quite different. Any engineer operating a saturated engine, especially with a long outside pipe line, notices the difference in the moisture of the steam in a good clear day, or a stormy day.

TABLE 3 RESULTS OF TESTS WITH SUPERHEATED STEAM IN BARE AND COVERED PIPES

Pressure, lb. per sq. in.	Final temp., deg. fahr.	Air temp., deg. fahr.	K B.t.u. per sq. ft. per deg. fahr.	Wall temp., deg. fahr.	B.t.u. per sq. ft. per deg. fahr.	Velocity, ft. per sec.	Temp. diff., steam and wall, deg. fahr.
93.8	540.8	50	0.70	530	31.8		10.8
92.5	549.0	50	0.72	540	39.8	68.3	9.0
93.8	551.2	50	0.72	544	49.8		7.2
93.8	556.0	51.8	3.79	493	30.4		63.0
96.7	567.0	51.8	3.85	513	36.7		54.0
98.2	579.8	51.8	3.9	524	36.7	98.5	55.8
95.3	586.2	51.8	3.9	525	34.0	Approx.	61.2
95.3	588.2	51.8	3.9	527	34.0		61.2
95.3	588.2	51.8	3.9	527	34.0		61.2
95.3	586.2	51.8	3.9	525	34.0		61.2
98.2	513.0	51.8	3.61	403	15.2		110.0
96.7	495.0	51.8	3.55	396	15.8		99.0
95.3	476.2	51.8	3.49	388	16.8		88.2
95.3	464.0	51.8	3.38	363	15.2	29.5 to	101.0
95.3	440.8	51.8	3.32	358	15.6	32.8	82.8
95.3	442.4	51.8	3.30	356	15.6		86.4
95.3	434.0	51.8	3.28	353	15.6		81.0
96.0	430.2	51.8	3.26	351	15.6		79.2
95.3	428.2	51.8	3.24	349	15.4		79.2

The Munich tests, to which the author refers, and with which the writer is quite familiar, were also made in a closed laboratory, and Eberle had recognized and admitted that the temperature difference between the steam and the wall, even with a covered pipe, increases considerably with the increased velocity of the air.

Particularly in discussing the heat exchange between the canvas surface and the air, the velocity and moisture of the air should not be neglected.

L. L. Barrett. The bare-pipe-loss curves and the canvas-surface-loss curves given in Figs. 2 and 4 represent a notable ad-

vance in the investigation of these subjects, and the author is to be congratulated on the results attained. Where results on different pipe sizes are obtained by the same investigator, these results are preferable to results obtained by other investigators each working on a single pipe size, as variables other than the one being considered are less likely to be present. By comparison of Figs. 2 and 4, it is noted that the heat loss per square foot per hour at 100 degrees temperature difference is greater from the 3.1-in. O.D. and 9.5-in. O.D. canvas-covered pipes than from the 3-in. and 10-in. bare iron pipes. This is at variance with previous results, which showed that the losses from bare pipes were greater than those from canvas-covered surfaces. The difference is understood to be accounted for by the fact that the author used thermocouples to obtain the canvas surface temperature whereas previous experimenters have used thermometers for the purpose.

The author's statement in Par. 18 that a 0.1-in. air space between the pipe and insulation increases the overall loss does not appear to be borne out by his further statement that the temperature drop for a 0.1-in. air space is approximately equal to that of 0.1 in. of insulation. If it is true that 0.1-in. air space is equivalent to 0.1 in. additional insulation, the overall loss is decreased by the use of the air space, assuming that there is no heat loss through the joints of the covering. However, the author's conclusion that coverings should be kept as close as possible to cylindrical surfaces is true for the reason that in commercial installations where coverings are not carefully sealed an air space will allow the circulation of convection currents along the pipe and the heated air will escape through the joints in the covering.

The curve for 1-in. pipe in this figure seems too far removed from the other curves for consistency. The matter can be investigated on the basis of the author's statement that the temperature drop through a 0.1-in. air space is approximately equal to that through 0.1 in. of insulation. The temperature drop through the first 0.1 in. of covering on a 1-in. pipe, assuming a standard thick 85 per cent magnesia covering and a temperature difference between pipe and room of 370 deg., figures out as 49 deg. which is one-third less than the corresponding ordinate on the author's curve for the 1-in. pipe. The temperature drops through the first 0.1 in. of insulation on 3-in. and 10-in. pipes figure out greater than the corresponding ordinates of the author's curves for these pipe sizes.

This makes it appear that the ordinates of the curve for 1-in. pipe are too great. Imperfect sealing of the ends or joint of the covering tested, thus allowing air infiltration, would account for the results obtained.

It is unfortunate that the author gives the impression in Par. 20 that conductivity is in some way related to the curvature of the covering. The definition of the thermal conductivity of a material is that it is the quantity of heat transmitted in unit time through unit area of a plate of unit thickness having unit difference of temperature between its faces. The proper conception of conductivity is therefore a conception of heat flow in one direction between two parallel planes. It is a specific property of the material and is independent of its shape. When heat flows through a material having curved surfaces, such as the insulating covering on a pipe, the increase of heat flow is accounted for by the increasing area of the path through which the heat may flow. The conductivity of the material, however, remains the same. In solving a problem of the heat flow through a pipe covering the correction for curvature is readily made, thus making it unnecessary to introduce the idea of curvature into the conception of conductivity.

In Par. 22, the author gives his ideas as to the direction in which future experimental work should proceed. This is a most important subject as manufacturers and users of insulations are constantly demanding more accurate data on conditions which are not directly covered by experimental research. To meet this demand various refinements in calculating processes have been introduced. One of these refinements which was introduced by G. D. Bagley,[1] was the plotting of conductivity curves as a function of the temperature difference of the two surfaces of the insulation. The author of the present paper apparently has in mind, in Par. 22, the plotting of some such curves for commercial pipe coverings, although he does not state whether these curves are to be plotted as a function of temperature difference or not. He also mentions plotting them for different thicknesses of covering. It therefore is desirable that this process of plotting conductivity as a function of temperature difference be examined to see whether it affords a rational basis for the calculations of heat losses. If at the same time we can find a method which eliminates the necessity for

[1] Conservation of Heat Losses from Pipes and Boilers, TRANS., vol. 40, 1918, p. 667.

plotting conductivity curves for every thickness of covering we shall have simplified matters considerably. Now we do know that conductivities change with temperature. If we are to accept Bagley's method, we must agree that the conductivity of a plate of material the two faces of which are at temperatures 200 and 100 deg. fahr. is the same as the conductivity of a plate of the same material the two faces of which are at temperatures of 600 and 500 deg. fahr. This does not seem logical and causes us to examine again the question of what conductivity is. We know that the conductivity of a substance is determined by the physical and chemical nature of the substance and that its physical and chemical nature is dependent upon its temperature and not upon a difference of temperature. The thermal conductivities of the metals and of some of the electric insulators have been determined by Lees, Hornbeck, and other physicists, all of whom have expressed the conductivities they obtained as conductivities at certain temperatures. Now if by experiment we establish our curve of conductivity as a function of the temperature (not temperature difference) and we find this curve to be approximately a straight line between the temperatures corresponding to the temperatures on the two surfaces of an insulating covering, we have the elegant and useful relationship that the equivalent conductivity for the whole thickness of the insulation is equal to the conductivity corresponding to the arithmetic mean of the two surface temperatures. The proof of this proposition is given by Hering.[1] In order that the proposition may be true, it is unnecessary that the conductivity be proportional to the absolute temperature as was assumed by Hering, but only that it be a linear function of the temperature such as $k = aT + b$, where k is the conductivity, T the temperature, and a and b are constants.

There is reason to believe that the conductivity curves of commercial insulating coverings when plotted against temperature will approximate sufficiently to straight lines to permit advantage being taken of the relationship referred to. The German physicist Nusselt[2] gives curves of conductivity as a function of temperature in Fig. 9. The curve for asbestos is a straight line between 200 and 600 deg. cent., while that for burned kieselguhr is a straight line from 0 to 450 deg. cent., and the curves for the other substances tested could be approximated by straight lines. The present knowl-

[1] *Trans.* Am. Electrochemical Soc., vol. XXI, p. 520.

[2] *Zeitschrift der Vereines Deutcher Ingenieure,* p. 1006, vol. 52.

edge of the relationship of conductivity and temperature has been summarized as follows by Pierce and Wilson:[1] "Most experimenters have been able to reproduce mathematically the results of their work on thermal conductivities by assuming that the conductivity is a linear function of the temperature."

There is therefore some ground for the conclusion that if we are to refine the methods for the computation of heat losses by taking into consideration the effect of temperature on conductivity, we should reject the conception of conductivity being dependent upon temperature difference and should establish instead the curves of conductivity as a function of temperature. In the experimental work incident to the establishing of these curves, it is desirable that the differences of temperature used should be comparatively small and that the mean temperature of the substance under test should be ascertained by means of temperature measurements at both surfaces. In order to keep the temperature difference between the two surfaces low when investigating the conductivity at the higher temperatures, it will be desirable to place an outer insulating covering over that which is being tested, as this will cut down the temperature drop in the covering under test. After such a curve is established, it is only necessary to enter the curve with the mean temperature of the two surfaces of an insulation to ascertain the equivalent conductivity of the insulation for use in the current formulae. This process is applicable whatever the thickness of the insulation, so that it becomes unnecessary to establish conductivity curves for each thickness of covering as outlined by the author in Par. 22. The result is to simplify greatly the experimental work required and to put the computation of heat losses on a more rational basis. Inasmuch as the author has not established conductivities in the experiments reported in this paper the suggestion here advanced does not detract in any way from the excellent results which have been accomplished.

L. B. McMillan. This paper is a timely contribution on a very important subject. It is of special interest to note that the author's results on losses per square foot of surface on various sizes of bare pipe show how very much small differences between the large and small pipes than those given by Paulding. For example, Paulding's curve for 16-in. pipe falls below the author's curve for

[1] Amer. Acad. Arts & Sciences, *Proc.*, vol. 34, p. 24.

18-in. pipe, while Paulding's curve for ½-in. pipe is considerably above the author's curve for that size. Paulding's curves show that at 500 deg. temperature difference the loss from a 16-in. pipe is 25 per cent less per square foot than from the same area of ½-in. pipe, while under the same conditions the author shows that the difference is only about half as great as Paulding's curves would indicate.

The writer is of the opinion that the author's curves are the more nearly correct in this respect, because, while it is certain that there is a tendency for the rate of heat loss to be higher on the smaller pipes than it is on the larger ones, this difference is not as great as it is often assumed to be. Furthermore, the difference between rates of losses from small and large pipes are small compared with those caused by varying rates of air circulation on the same pipe. In other words, while pipe size may account for a variation of 15 or 20 per cent, difference in air circulation may cause a variation of upwards of 100 per cent.

It will be noted that the author's curves are based on tests of three pipes of different size, and that the other five curves were obtained by interpolation and extrapolation. An explanation of how this was done would be of interest. This is particularly important, in view of the fact that the author's curves differ considerably in slope from those of Paulding and other recent investigators. In this connection, referring to Par. 8, it would be of interest to know if the slope of the curve from the second test was the same as that from the first. It is stated that the tests checked exactly the same at high temperatures, but it is not stated whether or not there was any variation at low temperatures.

Referring to Par. 15, the temperature drop does not depend only on resistance of the air space. It depends also on the amount of heat flowing across the air space and out from the insulation, just as voltage drop depends upon the electrical current, as well as resistance. Therefore, since temperature drop is equal to the product of thermal resistance (expressed in proper units) and heat flow, it is not clear that the temperature drop should decrease at the higher temperatures, as stated in this paragraph. While the resistance decreases at high temperatures, the heat flow increases, and if it increases more rapidly than the resistance decreases, the temperature drops at the higher temperatures should continue to increase in spite of the decrease in resistance.

Therefore, some other explanation of the lower temperature drops at higher temperatures may be required. One such possible explanation might be increased air leakage into and out of the air space at high temperatures.

Some such explanation is required for the curve for 1-in. pipe, because it is hardly possible that the insulating value of a 0.1-in. air space on a 1-in. pipe would be equal to one-third of the total insulating value of the insulation, unless that insulation were very inefficient. In order to check this point the kind and thickness of the insulation should be given.

In this connection, it may be stated that the kind and thicknesses of insulation should be given for each of the curves, for the thicker the insulation the lower will be the temperature drop through the air space.

The writer has made tests to check the author's results on temperature drop across 0.1-in. air spaces on 1-in. and 3-in. pipes. In these tests standard thick 85 per cent magnesia was applied in such manner as to provide air spaces of a uniform thickness of 0.1 in. around the entire circumference of the pipe. At a temperature difference between pipe and room of 235 deg. fahr. the temperature drop across the 0.1-in. air space on 3-in. pipe was 11 deg., which checks within 3 deg. of the author's curve for 3-in. pipe. However, at a temperature difference of 242 deg. fahr. between pipe and room the temperature drop across the 0.1-in. air space on 1-in. pipe was only 19 deg., while the author's curve for this size of pipe at the same temperature difference showed a temperature drop across the 0.1-in. of about 56 deg. Therefore, it would seem, as pointed out above, that the high values of temperature drop shown by the author's curve for 1-in. pipe must be accounted for by some factor other than the normal temperature drop across the air space.

It is not clear on what the conclusion in Par. 18 regarding the value of air space above 0.25 in. on flat surfaces is based, as the paper contains no record of results accomplished by various thicknesses of air space on flat surfaces.

The statements in the last two sentences in Par. 18 seem directly contradictory. If it is true that a 0.1-in. air space has the same resistance as 0.1 in. of insulation, then the heat transmission with the 0.1-in. air space would be less than without it, because surely the transmission through 1.1 in. insulation would be less than

that through 1 in., all other conditions being equal. Due to practical considerations, the writer does not advocate applying insulation over such air space, but the author's conclusion would seem to indicate that such a procedure would give slightly higher efficiency. He has shown that an air space as large as 1 in. increases the loss of a 3-in. pipe, due to the increased area, but that the small air space is just as good as that much additional insulation. It would be interesting to know just what thickness of air space would give the minimum loss, but this would be different for all different pipe sizes, increasing as pipe size increased, and in view of the practical difficulties in the way of applying insulation in this manner, it is doubtful if tests to determine this would be of practical value.

The first sentence in Par. 19 is open to the same comment as that made in connection with Par. 13, viz.: that temperature drop through the insulation depends upon the product of resistance, and heat flow, and not on the resistance alone.

Referring to Par. 20, the absolute conductivity of the material is a specific property of the material, just as density and specific heat are specific properties. Therefore, it is not dependent upon the size or shape any more than the density or specific heat is dependent upon these conditions. However, it does vary at different temperatures, but the effect of this variation may be determined without making tests on all thicknesses and all pipe sizes, as suggested by the author.

In this connection, the writer feels that Mr. Barrett's discussion has clarified the situation very materially. The use of what Mr. Barrett terms "equivalent conductivity" which is the absolute conductivity at the arithmetic mean of the temperature at the two surfaces of the insulation is far simpler than the determination of conductivity by tests of all thicknesses of insulation on all sizes of pipes. Mr. Barrett has shown that the proposed method may be proven to be mathematically correct where the absolute conductivity curve with respect to temperature is a straight line and the writer believes that such curves for most efficient insulating materials are either straight lines or vary slightly from straight lines.

The curves showing losses from canvas-covered surfaces of various dimensions are of considerable interest. However, the surface resistance for an efficient insulation 1 in. or greater in thick-

ness is usually less than one-quarter of the total resistance. Therefore, small variations in surface resistance will have little effect on the total result. Furthermore, small changes in air circulation will have much greater effect on surface resistance than will difference in pipe sizes.

The writer would like to ask the author what curve he would use for flat surfaces? It is evident that if the general equation [4], is applied to a flat surface, the temperature difference will be 272.5 deg., regardless of all other conditions. This is obviously impossible, therefore, it is doubtful if the author's general equation can be used very much beyond the limits of his actual experiments.

THE AUTHOR. The main object of this paper has been to present some of the latest findings on bare and covered pipes operating at the higher temperatures and to give empirical formulas developed from the results, which would enable engineers to solve more readily heat insulating problems. It is to be understood that lack of space has prevented elaborate details concerning fundamental data, all of which have been covered fairly well by other investigators.

However, since some of these points have been brought out in the discussion, the author will reply as briefly as possible to the most important questions.

Referring to the temperature drop from the outer surface of the pipe to the inner surface of the insulation, which has been discussed by Messrs. Barrett and McMillan, the results of these tests indicate that ordinarily the drop through a 0.1-in air space, or the space between the pipe and the inner surface of commercial insulations, is equal to the temperature drop through 0.1 in. of insulation, or the temperature drop is approximately the same as would be obtained if the insulation could be made to fit absolutely tight to the surface of the pipe. This depends upon the thickness of the insulation, the size of the pipe, temperature, etc. For strictly accurate calculations this temperature drop should be taken into account, especially for pipes 1 in. in diameter or less.

The drop in temperature at the higher temperatures, as shown in Fig. 3, can be attributed mainly to increased radiation loss, since the radiation loss increases more rapidly than the heat flow increases. This drop in temperature will vary under different conditions of temperature, thickness of air space, diameter of cylinder,

etc. In general, it may be said that the results of these tests on cylindrical surfaces compare favorably with the results obtained by the Bureau of Standards on flat surfaces.

The statement is not made in Par. 20 that the absolute conductivity depends upon the thickness and curvature of the covering. It is stated that the absolute conductivity curve depends upon the thickness and curvature of the covering.

If the temperature gradient curve from the inner to the outer surface of the insulation is obtained as the author has done by measuring the temperature at successive points out through the covering, and the flow of heat is obtained at the same time, the true absolute conductivity of the insulation at any radius or temperature can be calculated readily from the relation that the flow of heat H through unit area per unit of time is equal to the conductivity K multiplied by the temperature gradient, or $H = K\frac{dt}{dr}$.

It will be found upon plotting the absolute conductivity of the insulation so obtained against the distance from the inner surface of the covering that the resulting absolute conductivity curve will vary with the thickness and curvature of the insulation. These variations cause considerable difficulty in solving heat-flow problems.

In order to obtain more ready means of calculating heat-flow problems, the author has suggested in Par. 22 that tests should be conducted to obtain a mean conductivity coefficient for different thickness of coverings. This mean conductivity coefficient is the mean of the conductivities at different points through the radius of the covering as obtained from the temperature gradient curve. However, it is not necessary to determine the temperature gradient curve in order to determine the mean conductivity coefficient for the different thicknesses. This can be determined by the relation that $H = \frac{K(T_1 - T_2)}{r_1 \log_e \frac{r_2}{r_1}}$,

where K, the mean conductivity coefficient for the temperature range of the covering, is plotted against the mean temperature between the inner surface and the outer surface of the covering.

The use of what Mr. Barrett terms "equivalent conductivity" reduces the amount of experimental work considerably, as the absolute conductivity at any temperature can be obtained

from the experimentally determined temperature gradient curve, although it will be very difficult to determine the correct temperature gradient curve for some materials.

In order that the " equivalent conductivity " theory be correct, it is necessary that the conductivity be a linear function of the temperature. While this is true for most of the materials, great care should be taken that the law is not applied to materials which do not obey this law. In materials in which the flow of heat takes place mainly by conduction, the conductivity is a linear function of the temperature, but there are some materials in which the flow of heat takes place largely by radiation and convection currents. In these materials the conductivity coefficient which embraces the flow of heat by radiation, conduction, and convection, will not always follow a straight-line law.

In reply to Mr. McMillan's discussion of the bare-pipe-loss curves, the slope of the curve for the two tests on the 1-in. pipe was the same. A short time after this paper was prepared, a check test was run on the 3-in. pipe at a temperature of 762 deg. fahr. with the room temperature at 85.6 deg. fahr. The B.t.u. loss at this temperature, or 676.4 deg. fahr. temperature difference, was 6.51. If a correction is made for a room temperature of 81 deg. fahr., it will be found that this point falls almost exactly on the extended curve for the 3-in. pipe, thus indicating that the curves for the different diameter pipes are parallel throughout.

The general equation [4] is approximately accurate for diameters up to 2 ft., as stated in Par. 26.

The loss of heat per unit area from flat surfaces varies greatly with the size and position of the body. The loss from the surface in a horizontal position is entirely different for the same surface in a vertical position. Also, the loss is different for the same flat surface facing downward than for it facing upward.

For these reasons, it is unreasonable to expect that the curves for cylindrical surfaces could be extended to include flat surfaces. To express accurately the surface loss law from flat surfaces would probably require at least three rather complicated equations.

In regard to Mr. Broido's discussion, the loss of heat from the surface of bare pipes is absolutely independent of the nature of the steam in the pipes. The loss depends only on the temperature difference between the pipe surface and the surrounding air, the temperature of the air or surrounding objects, the nature of the pipe surface, diameter, position, etc.

The temperature drop from the steam to the outer surface of the pipe for superheated steam varies greatly under different conditions. In calculating the bare-pipe-loss table for the superheated steam, a drop of 10 deg. fahr. was assumed. Under certain conditions, the drop would probably be less than this, while in other conditions the drop would probably be much greater, as mentioned by Mr. Broido.

No. 1849

THE EVAPORATION OF A LIQUID INTO A GAS

By W. K. Lewis,[1] Cambridge, Mass.
Non-Member

The author investigates the mechanism of the evaporation of a liquid into a gas as applied to such processes as are found in gas scrubbers, humidifiers, dehumidifiers, water coolers, air driers, etc. He establishes the formula for calculating the humidity of air from wet- and dry-bulb thermometer readings, and shows that the coefficient of heat transfer divided by the coefficient of diffusion equals the humid heat of the gas.

A LARGE amount of work has been done on the evaporation of water into air at temperatures below the boiling point. The dynamic equilibrium corresponding to the evaporation of water into air counterbalanced by the flow of heat from the air into the water is the basis of wet-bulb thermometry, the most useful method of determining the humidity of air.[2] In 1886 Desmond Fitzgerald [3] pointed out that the rate of evaporation of water into air is a function of the difference in partial pressure between the moisture in equilibrium with the evaporating water and the actual moisture content of the air in contact with it. It is true that Fitzgerald did not assume the rate of evaporation linearly proportional to this difference, but added a small correction term proportional to the square of the difference. Barrows and Babb [4] made a large number of determinations of evaporation from the surface of Maine lakes, and while their experimental determinations were subject to a large

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[2] Leslie, Nicholson's Journal, vol. 3, p. 461.
August, Pogg. Ann., vol. 5, p. 69, 1825.
Apjohn, Trans. Royal Irish Acad., vol. 17, p. 275, 1834.
Weilenmann, Meteorol. Zeit., vol. 12, pp. 268 and 368, 1877.
Maxwell, Zeit. f. Meteorol., vol. 16, p. 177, 1881.
O. D. Chowolson, Traité de Physique, vol. 3, part 3, p. 807, 1911.

[3] Journal, Am. Soc. C. E., 1886.

[4] U. S. Dept. of Interior, Washington, D. C., Water Supply Bulletin No. 279.

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percentage variation, none the less their results substantiate this proportionality. More recently Willis H. Carrier[1] has shown that the rate of water evaporation is, within experimental error, proportional to partial-pressure difference. The following is an analysis of the mechanism of such evaporation.

PRELIMINARY ASSUMPTIONS

2 For purposes of formula derivation assume a long tunnel through which unsaturated air is flowing at constant mass velocity. The walls of this tunnel are non-conductors of heat. Along the bottom of the tunnel is placed a mat or wick permanently wet with water supplied from below as evaporation takes place above. The water is furnished to this wick at every point at a temperature exactly equal to that of the water on the upper surface of the wick at that point. There is no motion of the liquid water parallel to the axis of the tunnel. The mass velocity of the air over the water is constant and sufficiently low so that heat generated by friction may be neglected.

3 The unsaturated air entering this tunnel will become humidified in passing through it owing to the evaporation of water. In consequence, the temperature of the air will fall, and if the tunnel be sufficiently long, the water and air will ultimately come to equilibrium.

NOTATION

4 In the formula to be derived, the following notation is used:

A = Area of liquid in contact with gas

H = Absolute humidity of gas, parts by weight of vapor per part by weight of vapor-free gas

h = Surface coefficient of conductivity of heat between gas and liquid, B.t.u. per unit time per unit surface area per unit temperature difference

k' = Coefficient of diffusion, parts by weight of vapor diffused per unit time per unit area per unit absolute humidity difference

k = Coefficient of diffusion, parts by weight of vapor diffused per unit time per unit area per unit vapor pressure difference

p = Partial pressure of vapor

P = Total pressure of vapor and vapor-free gas (i.e., barometer)

[1] Am. Soc. Refrigerating Engineers, Journal, May 1916, vol. 2, no. 6, p. 25.

θ = Time
t = Temperature
r = Latent heat of vaporization
s = Humid heat[1], number of heat units necessary to change the temperature of unit weight of vapor-free gas, *plus the vapor it contains*, one degree
W = Weight of liquid evaporated.

INTERACTION OF WATER WITH AIR

5 Now consider for the moment the conditions at any given point along the length of this tunnel. At this point the temperature, absolute humidity,[2] and pressure of water vapor in the air will be represented by t, H and p. Since the apparatus is continuous in its operation these conditions will remain unchanged at this particular point, but will vary from point to point along the tunnel. The corresponding quantities representing the condition of the liquid water in contact with the air at this particular point are t_w, H_w and p_w.

6 The mechanism of interaction of the water with the air is as follows: There exists over the water what is equivalent to a stationary film of air,[3] which insulates the water from the main body of the air. Through this air film heat is diffusing from the air into the water and through the same film there is diffusing, in the opposite direction, the water vapor formed by evaporation on the surface of the liquid. This evaporation cools the surface of water, and, since it is available from no other source, heat must be supplied solely by diffusion from the air. The heat of vaporization must therefore be quantitatively compensated by the heat flow

[1] W. H. Grosvenor, Proc. Am. Inst. Chem. Engrs., vol. 1, 1908.

[2] See notation, par. 4, and bibliographical references. Humidity is expressed as ratio of vapor to dry air because this ratio is unchanged by variations in either temperature or total pressure, and because differences in humidity directly represent loss or gain of vapor in the air.

[3] The existence of a stationary liquid or gas film at the point of contact with solid surfaces and its low heat conductivity has long been appreciated. The first mention of a water film was made by Péclet in 1844 (Traité de la Chaleur-Liège, chap. 8, p. 131) and since that date many investigators have checked this idea and demonstrated the existence of the film. The most tangible conception of the gas film is represented by Langmuir (Physical Review, vol. 34, 1912, p. 421) although the film concept was not his originally. The explanation in terms of the film concept of the effect of velocity upon heat transfer is definitely stated and the part played by velocity in tearing down this film noted.

through the surface film, and the rate of evaporation is limited by the rate of diffusion of vapor through the same film.

7 From the foregoing one can immediately write the following equations:

$$-\frac{dW}{Ad\theta} = k'(p_w - p) \quad \ldots \quad [1]$$

$$\frac{dQ}{Ad\theta} = h(t - t_w) \quad \ldots \quad [2]$$

$$dQ = -r_w dW \quad \ldots \quad [3]$$

Whence

$$p_w - p = \frac{h}{k'r_w}(t - t_w) \quad \ldots \quad [4]$$

This last equation is the one normally used for calculating the humidity of air from wet- and dry-bulb thermometer readings.[1] In it, variation in r_w is neglected and the term $\frac{h}{k'r_w}$ is assumed constant. For p in millimeters of mercury and t in deg. cent., it equals 0.5. The equation implicitly assumes that the cooling of the air is differential, i.e., so small in the neighborhood of the point in question that the actual changes in temperature and humidity of the air, t and p (or H), are negligible.

Since[2]

$$p = p\frac{\frac{H}{18}}{\frac{H}{18} + \frac{1}{29}} \quad \ldots \quad [5]$$

$$t - t_w = \frac{k'r_w}{h} P\left(\frac{\frac{H_w}{18}}{\frac{H_w}{18} + \frac{1}{29}} - \frac{\frac{H}{18}}{\frac{H}{18} + \frac{1}{29}}\right) \quad \ldots \quad [6]$$

Where H is small, as is usually the case below 150 deg. fahr., $\frac{H_w}{18}$

[1] Smithsonian Meteorological Tables, 1897, Introduction, p. xxxviii:

$$f = f_1 - 0.000367\, B(t - t_1)[1 + 0.00064(t - t_1)]$$

Physikalish-chemische Tabellen, Landolt-Bornstein, Psychrometertafel, p. 370:

$$a = m_f - \frac{1}{2}(t - f)\frac{b}{755}$$

[2] This involves the assumption, entirely valid under normal conditions, that the water vapor follows the gas laws. The deviations from the gas laws become progressively less as pressure decreases; even at saturated pressure at 150 deg. fahr. the deviation is less than 0.7 per cent. (See Steam Tables, Marks and Davis, Longmans, Green & Co., 1913.)

and $\frac{H}{18}$ are negligible compared with $\frac{1}{29}$, and one may write, as a close approximation,

$$t - t_w = \frac{kr_w}{h}(H_w - H) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where

$$k = 29k' \frac{P}{18}$$

8 It is obvious that h and k depend on the thickness of the air film and are therefore functions of the velocity of the air. It is, however, equally obvious that, if air velocity be increased sufficiently to double the one, the other will double also. The ratio of $\frac{h}{k}$ therefore remains constant, independent of velocity. This explains why the reading of a wet-bulb thermometer is uninfluenced by the velocity of air passing it, provided the velocity is sufficient so that any heat lost by radiation is negligible in comparison with that picked up by conduction.

THE RELATION $\frac{h}{k} = s$

9 Now consider the change in humidity and temperature of the air as it moves along the tunnel. Starting at the same point previously considered, the air will drop in temperature by an amount dt and increase in humidity by an amount dH. The heat given up by cooling must correspond to the heat of vaporization of the water picked up, i.e.,

$$-sdt = r_w\, dH \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

whence, assuming constancy of s and r_w,

$$H = -\frac{s}{r_w}t + \text{const.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

Assuming the tunnel indefinitely long, the air will ultimately become saturated at some temperature t_e, and humidity H_e. Since these conditions represent equilibrium between the air and the water, evaporation will cease, and t_e and H_e are therefore the constant, fixed end-points of the process. Inserting these limits,

$$H_e - H = \frac{s}{r_w}(t - t_e)$$

We have now derived two formulas connecting H and t, both applying to this same process of evaporation, i.e.,

$$H_e - H = \frac{s}{r_w}(t - t_e)$$

and

$$H_w - H = \frac{h}{kr_w}(t - t_w)$$

These two expressions must therefore be identical. By the method of undetermined coefficients this can be true only provided the corresponding coefficients are equal, i.e., $\frac{s}{r_w} = \frac{h}{kr_w}$, or $s = \frac{h}{k}$; and $t_w = t_e$, a constant;[1] and $H_w = H_e$, also constant.

10 So far the discussion has been limited to water and air. Obviously, however, the same relationships must apply to any liquid and any gas with which its vapor is mixed.

11 The first of these equations, $\frac{h}{k} = s$, states that the coefficient of heat transfer divided by the coefficient of vapor diffusion through the gas film is constant, and equal to the humid heat[2] of the gas. By means of formula [4] the ratio $\frac{h}{k}$ can be calculated from the observed wet- and dry-bulb temperatures for any vapor-gas mixture of a known gas humidity. The experimental determinations of wet-bulb temperatures for

a water-air
b water-carbon dioxide
c toluol-air
d chlorbenzol-air

and calculations for $\frac{h}{k}$ given in Table 1 show that this ratio is in all cases substantially equal to the humid heat of the entering gas, which in this case was identical with the specific heat because the gas used was vapor-free.

[1] W. H. Carrier (Journal Am. Soc. M. E., 1911, p. 1309) states as a general principle that the wet-bulb temperature of air remains constant during "adiabatic" evaporation of moisture into it. He gives, however, no proof or adequate explanation of the statement. From the above it is obvious that the statement is true only where the amount of vapor in the air is small, i.e., this relationship represents merely a limiting case. With high vapor concentration the wet-bulb temperature changes during such a process. The variation, however, is negligible for almost all engineering work.

[2] W. H. Grosvenor, Proc., Am. Inst. Chem. Engrs., vol. 1, 1908.

TABLE 1 VALUES OF $\frac{h}{k}$ FOR VARIOUS VAPOR-GAS MIXTURES

	$\frac{h}{k}$ Calculated from Experimental Results	Specific Heat of Gas
(a) Water-air	0.236	0.238
(b) Water-carbon dioxide	0.217	0.220
(c) Toluol-air	0.238	0.238
(d) Chlorbenzol-air	0.248	0.238

12 We have therefore demonstrated that, granting substantial constancy of s and r_w, and assuming H to be small, the ratio of the coefficient of diffusion of heat to that of any vapor through the gas film on the surface of the liquid is equal to the "humid" heat of the gas. Furthermore, during "adiabatic"[1] evaporation of a liquid into a gas, the liquid being in dynamic equilibrium with the gas, the temperature of the liquid remains unchanged throughout the process and the end-point of the process is reached when the gas has cooled itself to saturation at a temperature identical with that of an ordinary wet-bulb thermometer.[2]

IMPORTANCE OF THE RELATIONSHIP $\frac{h}{k} = s$

13 The importance of the relationship $\frac{h}{k} = s$ is very great. The term s, the humid heat, may be readily calculated for any case, regardless of whether the problem is primarily one of heat transfer or of diffusion. Hence if the heat-transfer coefficient h has been experimentally determined for a certain type of apparatus operating under definite conditions, the coefficient of diffusion equals $\frac{h}{s}$, and the capacity of this same apparatus may be predicted when functioning in diffusion processes, e.g., as a gas scrubber. Conversely, if k and s are known for definite conditions, h equals ks; in other words, one can predict the performance of a given apparatus for heat transfer from data upon the same equipment functioning as a scrubber.

14 These processes of diffusion of heat and of vapor are at the basis of the performance of all such equipment as humidifiers, dehumidifiers, water coolers, gas scrubbers, air driers, light oil stripping

[1] W. H. Carrier, Journal Am. Soc. M. E, 1912, p. 1321.
[2] Ibid.

columns, and the like. The above relationships make it possible to study the performance of such equipment on a more rational basis than hitherto and to compare the effectiveness of different types of equipment even when the data on the individual types are obtained under widely varying conditions. The Department of Chemical Engineering, Massachusetts Institute of Technology, expects to publish in the near future a series of articles showing various applications of these relations.

DISCUSSION

W. H. Carrier. The writer is gratified to have the further confirmation of the psychrometric relationships and theories, which he endeavored to establish in his paper on Rational Psychrometric Formulae,[1] to which Professor Lewis refers. In this paper, the writer approached this subject from the standpoint of the heat balance and experimental confirmation of the identity of the wet-bulb temperature with that of adiabatic saturation.

The author approaches this from the heat flow standpoint, arriving at identical results, and giving perhaps an added confirmation of the truth of the statement, or theory, that the wet-bulb temperature is constant so long as the total heat is constant. In other words, that the wet-bulb temperature is constant during adiabatic saturation, and that the final temperature of adiabatic saturation is the same as the wet-bulb temperature, which is the temperature of the liquid, during evaporation.

In his reference to the writer's paper, the author remarks that this statement was made, but that no proof was given. The author is referred to Par. 24, page 1014, of the original paper, which reads as follows: "A statement of experimental demonstration of these four principles is given in Appendix 1, Par. 67, pages 1032–1033."

Reference should also be made to Appendix 3 of this same paper, pages 1037–1039, showing two methods of derivation of the formula, and defining the relationship between sensible and latent heat in the adiabatic change. The derivation of this re-

[1] Trans., vol. 33, 1911, p. 1005.

lationship was fully discussed by Prof. G. A. Goodenough, pages 1041–1044.

The author further states in his note to Par. 9: "from the above it is obvious that the statement (that the wet-bulb temperature is constant during adiabatic saturation) is true only where the amount of vapor in the air is small, that is, the relationship merely represents a limiting case. With high vapor concentration, the wet-bulb temperature changes during such a process."

Assuming the correctness of the statement that the true wet-bulb temperature (i.e., disregarding small radiation error) is equivalent to the temperature of adiabatic saturation, this conclusion by the author is manifestly impossible. At the temperature of adiabatic saturation, the wet-bulb temperature, of course, is precisely the same as the temperature of adiabatic saturation, since wet- and dry-bulb temperatures and dew point are identical at this point. Also, (and this is important), if it is true that the wet-bulb temperature corresponds to the temperature of adiabatic saturation for any condition, then at any point during the process of adiabatic saturation, the wet-bulb temperature must remain the same, because it is manifestly impossible to change the temperature of final adiabatic saturation by any adiabatic process. It is then seen that Professor Lewis' statement amounts to a denial of the fact that the true wet-bulb temperature corresponds to the temperature of adiabatic saturation.

It is not possible to answer this question by any mathematical analysis, except through certain assumptions, as for example, those made in the author's paper, with reference to the action of the interchange of heat at the surface film. It can, however, be demonstrated experimentally, and the writer has given the method of such a demonstration in Appendix 1, Trans. A.S.M.E., Vol. 33, 1911. Further experiments are at present being conducted at Cornell University, employing methods of great precision and covering a wide range of temperatures and humidities, and all the data so far obtained confirm this identity within the range of experimental error (that is, about 0.05 deg. fahr., regardless of the moisture content.

Furthermore, if there were any variation in the wet-bulb temperature, the greatest variation would not occur, as the author

remarks, at high moisture content, but at high temperatures and low moisture content. If we assume this to be correct with low moisture content, as, for example, at zero moisture content, then mathematically and physically it must follow that it is true for all other values. This was mathematically demonstrated in Appendix 3, pages 1037–1039, of the writer's paper. The mathematical and physical proof was thoroughly discussed on pages 1041–1044, by so eminent an authority on this subject as Prof. G. A. Goodenough of the University of Illinois, and the conclusions which the writer reached were definitely verified, so that it must either be proved that the true wet-bulb temperature identity with the temperature of saturation does not hold at low moisture content, regardless of temperature, or else it must be proved that the physical and mathematical proof given by the writer and Professor Goodenough, for the equation of the adiabatic saturation line, is not correct.

The writer's paper on the Theory of Evaporation in connection with Compartment Driers, presented to the American Chemical Society[1] may also be cited as a reference. In this it was shown that the rate of heat transfer by evaporation, as determined by the difference in vapor pressures, is approximately equivalent to the heat transfer corresponding to the difference in the wet- and dry-bulb temperatures, or, in other words, that the latent heat transfer is approximately proportional to the sensible heat transfer, so far as rate is concerned, one being proportional to the difference in vapor pressures, and the other proportional to the difference in temperatures; these differences being related through any of the various psychrometric formulae.

This, however, does not detract in any way from the ingenious and admirable method presented by Professor Lewis. There is one point in his demonstration, however, which is not quite conclusive, apart from experimental evidence, and that is the assumption that the rate of latent heat and moisture diffusion follows the same law with respect to change in air velocity, as does the law of sensible heat transfer, with corresponding changes of velocity. No matter what method of attack is employed we return to the experimental evidence that the wet-bulb temperature is identical with the temperature of adiabatic satura-

[1] *Journal Industrial and Engineering Chemistry*, vol. 13, no. 5, May 1921.

tion, and that the rate of heat transfer is substantially proportional to temperature difference, and, also, that they both vary approximately according to the same law, over certain ranges of velocity. Present data would seem to indicate that the rate of moisture "diffusion" for a given vapor pressure difference has a certain minimum at zero velocity, and increases in direct ratio with increase of velocity, while all experimental evidence that we have at present would indicate that the rate of sensible heat transfer does so only approximately, but falls off gradually as the velocity increases. The experimental evidence on evaporation cannot be said, however, to be conclusive in this respect, at present, and is apparently contradicted by experiments with the wet- and dry-bulb "Kata-thermometer," which have shown, with the bulb dry, that the heat transfer occurs in proportion to the difference in sensible heat between the air and that of the film, at all velocities, while with the wet "Kata-thermometer," the rate of heat transfer occurs in exact proportion to the difference of total heat, that is, sensible and latent, between that of the circulating air and that of the surface film. The latter experiment would indicate that convection and radiation occur in uniform proportions for all velocities. These experiments were conducted with both vapor pressures and temperatures considerably above those of the surrounding air, and over a wide range of variations in temperature and velocity, so that they seemed to be quite conclusive on this point. These experiments were made at the Research Laboratory of the American Society of Heating and Ventilating Engineers in Pittsburgh, and are discussed in a paper by O. W. Armspach, *Journal* A.S.H.&V.E., January 1922.

In his analysis of the process occurring in the interchange of heat, the author speaks of the heat as being transferred through the gas film to the water. On this assumption there would, of necessity, be a difference in temperature between the film and the water. Otherwise heat transfer could not occur. We would then be forced to the assumption that this film was not saturated, or else if it were saturated, that the temperature of the water on the bulb was below that of adiabatic saturation in the film. As a matter of fact, it seems probable that when the temperature of the water on the bulb has reached its minimum, there is a condition of stability reached in which no heat

is transferred, either to or from the water on the bulb, (except a small amount from radiation), and that by the purely convection process, the particle of air impinging into the film becoming saturated and giving up its corresponding sensible heat simultaneously at the film surface.

There is other evidence which apparently disproves this to a certain extent. At very low velocities there is apparently heat conducted through the film to the bulb in excess of the rate at which the moisture is diffused, for it has been shown by the writer that the radiation error increases rapidly as the velocity decreases. In still air this discrepancy is about 17 per cent. (See Radiation Error in Wet Bulb of Sling Psychrometer, Rational Psychrometric Formulae, TRANS. A.S.M.E., Vol. 33, pp. 1025). This error between the theoretical and actual depression at low velocities is considerably greater than anything that can be accounted for by pure radiation alone. Therefore it must be due to conduction, which indicates that the rate of heat transmission is greater than the rate of vapor diffusion at low velocities, and that these rates gradually approach the same value as the velocity is increased.

The author's use of the term "diffusion" seems to differ somewhat from the ordinary conception of that expression. Under artificially produced velocities, the interchange of heat and moisture occurs through a process of *forced mechanical mixture,* rather than what is ordinarily understood by the term "diffusion," which, in the case of a vapor, is more analogous to the conduction of heat in still air. The laws of gas diffusion or mixture and the corresponding convection of heat can only be determined empirically by experiment, in which there is considerable opportunity for slight experimental errors, and for considerably divergent results through slight variations in experimental method, so that any psychrometric relationship based upon the comparison of these experimental rates must always be considered only as approximate, and useful for engineering purposes only, rather than of scientific value.

For industrial engineering purposes, at any except very low velocities, the approximate relationship given by the author may be said to hold with sufficient accuracy. For scientific work, however, the other variables, such as radiation and transmission, must be taken into account, as must also the variation of the

psychrometric coefficient with the wet-bulb temperature. The formulae given in the author's paper for determination of the vapor pressure in wet- and dry-bulb temperature readings are not accurate, as they neglect this factor, and assume the vapor pressure differences proportional to the depression, while the experimental work done by the Weather Bureau (Annual Report, Ch. Signal Officer, 1886, Appendix 24, pp. 233-259) shows conclusively that the relation $\frac{e' - e}{t - t'}$ varies with the wet-bulb temperature. The same thing is also shown by the mathematical and physical deduction of the writer.[1]

The author also compares the values of h/k as calculated from experiments made upon the temperature of evaporation, with the specific heat of the gases. Attention should be called to the fact that the apparent close agreement between the experimental values of h/k and the specific heat of the gases is misleading. In the first place the experimental data upon which are based the values of h/k are neither shown nor quoted. Determination of the specific heat by the evaporative method is not at all an easy task, and requires very special equipment and very accurate instruments for determination and control of conditions, in order to eliminate the radiation and conduction errors and to determine values with the degree of exactness given by the author. In the second place, the value used by the author for the specific heat of air (0.238) is now considered obsolete. Probably a more accurate value was that given first by W. F. G. Swann, *Phil. Trans. Royal Soc.*, series A., vol. 210, pp. 199–238, 1909, which was used by the writer in 1911, because it conformed almost exactly with the experimental determinations as shown by the data of the U. S. Weather Bureau, of wet-bulb and dew-point temperature relationships. This value is

$$C_{pa} = 0.24112 + 0.000009t \text{ deg. fahr.}$$

A still later experimental value was determined in 1913 at the Reichsanstalt by the thorough investigations of Holborn and Jakob (*Zeitschrift des Vereines Deutscher Ingenieure,* vol. 58, p. 1436), which gives the mean value of C_{pa} as 0.2415 between the

[1] Trans. vol. 33, 1911, pp. 1023–1024, also Theory of Atmospheric Evaporation with Special Reference to Compartment Dryers, *Journal of Industrial and Engineering Chemistry,* vol. 13, no. 5, equation 6.

limits of 68 and 212 deg. fahr., therefore almost exactly confirming the values obtained earlier by Swann, so that this value at present should be used rather than the old value of 0.2375, obtained by Regnault, which is apparently the value used by the author. If the present accepted value is used, then the comparison between the values of specific heat as given is not nearly so close. Furthermore, the only published data on evaporative temperatures corresponding to different vapor pressures and air temperatures are those given by the U. S. Weather Bureau. Their data also agree with this value rather than the lower one, although there are probably discrepancies in the U. S. Weather Bureau data, which must be taken into account. These probable errors in the U. S. Weather Bureau data seem very largely to neutralize each other, insofar as the effect on specific heat is concerned. There is one experimental error in their determinations of the dew points which gives somewhat higher values than the actual. The possibility of this error was pointed out by the writer (Trans. A.S.M.E. vol. 33, 1911, Par. 11 and 12, p. 1010). Offsetting this there is apparently an error in the specific weight of vapor in mixture with air. Dalton's law does not seem to hold with accuracy for mixtures of gases and vapors. The possibility of this error was pointed out by the writer in his closure page 1052, Trans. A.S.M.E. vol. 33, 1911.

Experiments are now being conducted under the writer's supervision at Cornell University, which will show the sources and amounts of these errors. Therefore present data on the weight of any vapor per cubic foot in a mixture of air or gas cannot be used with entire assurance. The source of this error may be clearly shown by theoretical thermodynamic calculations, as well as by experiments. On account of these errors in accepted data, no conclusions with reference to the agreement of experimental values of h/k and data on specific heat do not seem warranted. The only conclusion from experimental data that is warranted is the substantial identity of the wet-bulb temperature with the temperature of evaporation, except for a slight radiation and conduction error at high velocities.

The Author. The major point made by Mr. Carrier is his disagreement with the conclusion reached by the author that the true wet-bulb temperature does not quantitatively correspond to the temperature of adiabatic saturation. It is true that "it

is not possible to answer this question by any mathematical analysis," and also true that the vapor pressure of water at ordinary temperatures is so slight that the difference between the observed wet-bulb temperature and the point of adiabatic saturation is less than the experimental error. This laboratory is endeavoring to secure dependable experimental data on the point, using liquids of higher vapor pressure. Until such data are obtained discussion of this particular point will not prove conclusive.

Mr. Carrier states "it seems probable that when the temperature of the water on the bulb has reached its minimum, there is a condition of stability reached, in which no heat is transferred, either to or from the water on the bulb, (except a small amount from radiation.)." Observation shows two facts, first, that under these conditions the water on the bulb is evaporating, and second, that during this evaporation the temperature of the bulb remains substantially constant. Where then does the heat of evaporation of the water come from?

As regards experimental methods used by the author, the liquid was placed in a Dewar bulb and air or CO_2 dried by chemical absorption forced through it. The bulb was immersed in a liquid bath artificially maintained at the same temperature as the liquid within the bulb, so that radiation losses to the outside were thereby eliminated. The amount of contact of the gas with the liquid was controlled by varying the depth of the latter and by the use of solid filling material in the bulb. The only serious source of error encountered was the cooling of the entering gas by heat conduction along the tube through which it entered before it came into contact with the liquid. To reduce this error the tube glass was vacuum jacketed. With these precautions the wet-bulb reading was found independent of air velocity and of amount of contact with the liquid in the cases reported but not for liquids of high vapor pressure. We believe this latter discrepancy is partly caused by lack of constancy of the wet-bulb temperature, but the experimental data are not yet sufficiently refined to prove it.

It will be noted that the average of the calculated values for the specific heat of air is 0.241. This checks better with the modern data than with the Regnault figure, contrary to the statement of Mr. Carrier.

Further confirmation of the relationship $h/k = s$ and illustrations of its engineering usefulness will be found in an article by Whitman and Keats [1] and in one to appear later in this journal by Professor Robinson.

[1] Rates of Absorption and Heat Transfer between Gases and Liquids, *Jour. Ind. Eng. Chem.*, vol. 14, 1922, p. 186.

No. 1850

POWER DEVELOPMENT IN THE SOUTHEAST

By Charles G. Adsit,[1] Atlanta, Ga.

Non-Member

IN considering the power development in the Southeast, it is perhaps in order to give a brief history of the development and use of hydroelectric power since this form of energy predominates in this section.

2 It has been found that some form of water wheel was in use with primitive peoples as far back as any record is available, the first ones raising water for irrigation and for grinding grain. Water wheels have gone slowly through the various steps of development from these early times, but it was not until their application to driving electric generators that improvement of design and efficiency began. Since that time, they have grown from a few horsepower to 50,000-hp. units. The serious construction involved in the application to hydroelectric power began in the early 90's, but at that time its use was still restricted by the limitation of transmission.

3 In 1897, the highest voltage of transmission was 20,000; in 1900 this had grown to 40,000; in 1903 to 60,000; in 1908 to 100,000; in 1913 to 150,000; and today there are two lines being built in California whose operating voltage will be 220,000. One of these lines is approximately 250 miles long, the other something more than 300 miles.

4 The increase in voltage has made available many hydroelectric developments which in former years were too far from the center of application for the power to be transmitted, since the location of hydroelectric possibilities are not usually in sections which lend themselves to a large growth of population.

[1] Director and Executive Engineer, Georgia Railway & Power Co.

Presented at the Spring Meeting, Altanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

5 As the use of electric power became more and more general, and was not confined merely to lighting, the improvement and increase in size of generators became rapid, and the ease with which electric current could be applied to lighting and power purposes made its application imperative throughout a very wide field, at present embracing some 3000 distinct uses.

6 It has been said that the growth of any community — large or small — depends largely on two factors, transportation and available power. It is therefore of the greatest benefit to any community to do everything possible to promote power developments and to show its public utility companies the greatest consideration.

POWER RESOURCES OF THE SOUTHEAST

7 The southeastern section is traversed by the great Appalachian Mountain Range, which rises to an altitude of from one to seven thousand feet above sea level, and enjoys an average annual precipitation greater than that of any section of the United States, the average ranging from 50 in. to 85 in. per annum. In the foothills of this mountain range are many streams and rivers which have a fall, as a rule, too great for them to be navigable and so lend themselves admirably to economical development. This region also enjoys a climate of even temperature not subject to severe winters or ice conditions. The mean annual temperature varies from 55 deg. at an elevation of 2000 ft. above sea level to about 60 deg. at 500 ft. elevation.

8 The region is somewhat heavily timbered, especially on the head waters of the streams, and this to a large extent prevents a sudden run-off or great floods except during periods of extraordinary precipitation.

FIVE-STATE DEVELOPMENT

9 It is intended to deal herein largely with the water-power and steam-generating stations that are now connected to the transmission system covering the five states of Alabama, Tennessee, Georgia, and North and South Carolina. The transmission system as shown in the accompanying map, Fig. 1, includes the five states mentioned, forming a trunk line some 900 miles long with several thousand miles of main line connected. It is operated at 110,000 volts, 3-phase, and at 60-cycle frequency. All of the large

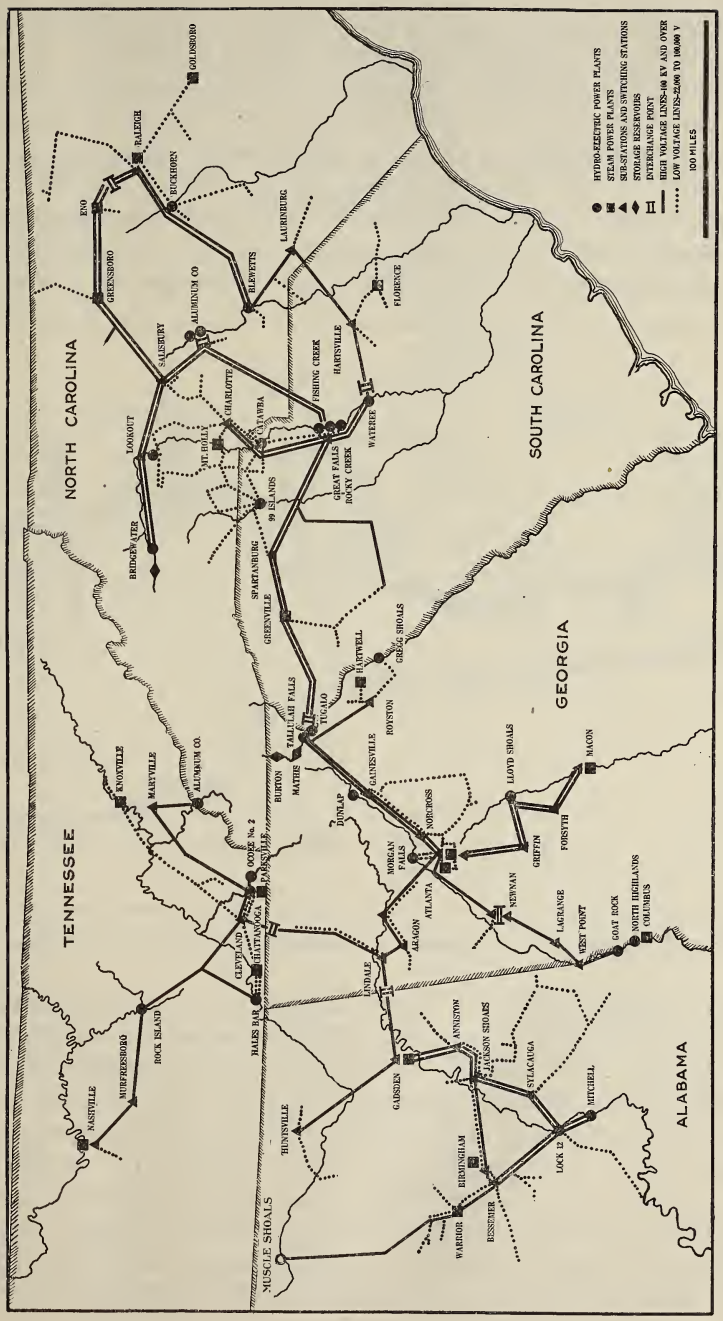

Fig. 1 Five-State Transmission System

water-power and steam-generating stations in this territory are connected to this transmission system, making it possible to transfer power from one locality to another. This relieves power shortages due to periods of drought at water-power stations, or shortage of coal at steam plants or any other unforeseen circumstances which may interfere with the power supply in the different sections. In several instances, every power station connected to this system has been operated in parallel at the same time and with marked success. During the month of October 1921, a part of Georgia and a part of

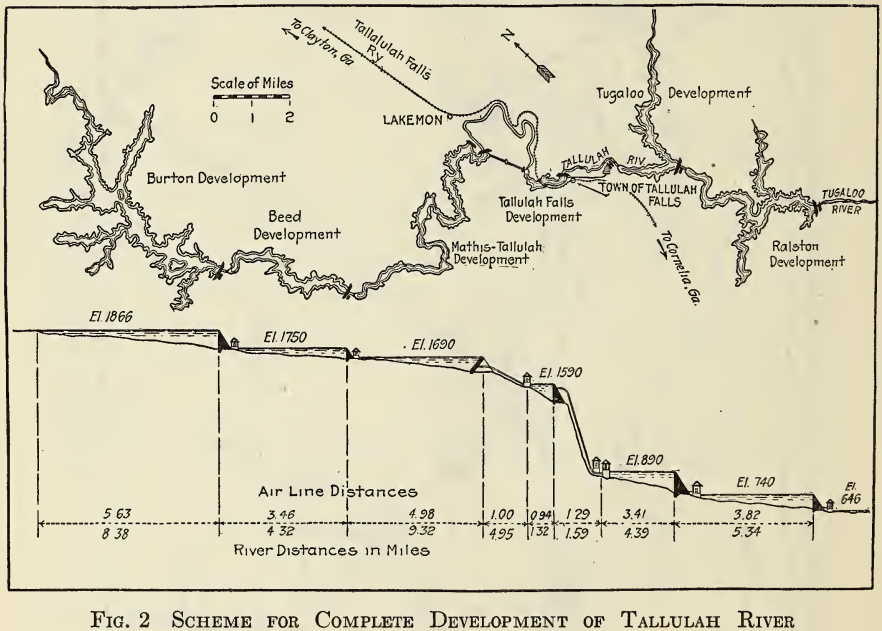

FIG. 2 SCHEME FOR COMPLETE DEVELOPMENT OF TALLULAH RIVER

North Carolina suffered a severe shortage in water due largely to the fact that the precipitation for the preceeding six months was considerably below normal, but by means of the existing transmission system, both the affected areas were fully supplied with power from other parts of the system. Had this great transmission system not been in service, and interconnected, the industrial load in these two localities would have been necessarily dropped, since the power companies could not have supplied a larger amount of power than that required by the municipalities which they also serve. However, the other power companies by lending their assist-

ance prevented any shutdown whatsoever in the industrial plants and the consequent unemployment of many thousand operators.

10 The following table gives the installed capacity of the various power developments which are connected to the system

POWER-GENERATING STATIONS OF THE POWER COMPANIES OF THE SOUTHEASTERN SECTION

Hydroelectric Stations		Steam Stations	
Alabama Power Co.			
	kw.		kw.
Lock 12	72,000	Warrior	60,000
		Gadsden	10,000
		Birmingham	10,000
Tennessee Power Co.			
	kw.		kw.
Parksville	15,000	Parksville	6,125
Hales Bar	37,100	Nashville	15,500
Great Falls	7,800	Knoxville	6,125
Ocoee No. 2	10,500	Chattanooga	3,000
Georgia Railway & Power Co.			
	kw.		kw.
Tallulah Falls	72,000	Atlanta No. 1	11,000
Morgan Falls	10,500	Atlanta No. 2	10,500
Dunlap	2,200		
Chestatee	1,100		
Gregg Shoals	1,200		
Columbus Power Co.			
	kw.		kw.
Goat Rock	11,000	Columbus, Ga.	9,000
North Highlands	6,900		
Central Georgia Power Co.			
	kw.		kw.
Lloyd Shoals	14,400	Macon Nos. 1 and 2 each	1,500
Southern Power Co.			
	kw.		kw.
Bridgewater	20,000	Eno	10,000
Lookout	18,000	Greensboro	6,800
Catawba	4,800	Mount Holly	6,800
Fishing Creek	30,000	Greenville	6,800
Great Falls	24,000		
Rocky Creek	24,000		
Wateree	56,000		
99 Islands	18,000		
Carolina Power & Light Co.			
	kw.		kw.
Buckhorn	2,900	Florence	1,500
Blewetts	33,100	Raleigh	3,720
		Goldsboro	1,200
Total kw.	492,500	Total kw.	181,070
Total hp.	738,750	Total hp.	271,605
Total developed horsepower, hydroelectric and steam			1,010,355

under discussion. From this table it will be noted that the total installed capacity is 1,010,355 hp. The total output of the system during 1921 was approximately two billion kilowatt-hours.

11 New developments are constantly being made by the various power companies comprising this system, and there is now under construction an additional 310,000-hp. capacity in generating equipment. This does not include 90,000 hp. which will also be installed as an addition to one of the new plants at a later date, nor the much-discussed Muscle Shoals hydroelectric development, mention of which will be made later.

WATER-POWER RESERVES

12 The steam-generating stations of the Southeast, generally speaking, are used for the purpose of supplementing the water-power production. Many of the water-power plants are, however, also backed by large storage reservoirs, making their annual output practically continuous regardless of dry months or dry years. On the head waters of the Catawba River, the Southern Power Company has constructed a very large storage reservoir at Bridgewater, N. C., which has a capacity of 13,500,000,000 cu. ft., representing 90,000,000 kw-hr. under present conditions at the various water-power plants already constructed below. The Georgia Railway & Power Company has two storage reservoirs on the Tallulah River, one at Burton containing 5,280,000,000 cu. ft. and one at Mathis containing 1,369,000,000 cu. ft. This total storage on the Tallulah River represents approximately 70,000,000 kw-hr. at the present developments and makes available for use in power production at the plants below, about 95 per cent of the annual stream flow. This storage will later be equivalent to 125,000,000 kw-hr., when other developments contemplated and under construction are completed.

13 The diagram given in Fig. 2 shows the scheme for the complete development of the Tallulah River with reservoirs and power plants. These developments when completed will have an average annual output of 535,000,000 kw-hr., the reservoir capacity falling through 1220 ft. head. In this section the average annual precipitation is about 65 in., and the average annual run-off about 40 in.

14 The entire section of the Southeast is fortunate in having water power available which can be depended upon so completely

due to the fact that great droughts seldom occur. There is no section of the United States with the exception of that adjacent to Niagara Falls, or in sections where the streams are fed from perpetual snows, where the flow of the streams and rivers can be so thoroughly depended upon. This insures a continuous supply of power to the various industries located in this section, with very little secondary power available or desired.

15 It may be interesting in passing to note that the power connected to the transmission system under discussion represents 17.75 per cent of the total water-power development in the United States.

UNDEVELOPED RESOURCES

16 While the Southeast has enjoyed extensive hydroelectric development already, it still has many sources of potential water power available for development as rapidly as the demand for electric power increases, and it will be many years before the entire possibilities of the streams in this section are exhausted. There are many industries not supplied at present with hydroelectric power which are depending upon the cheapness of fuel in the section where they are operating, and which in time will turn to electrical energy for their power needs. The electrification of the railroads of the South, which cannot be put off for many more years, will also provide an outlet for a large block of electric power, many of the trunk train lines being already paralleled by transmission lines of sufficient capacity to meet their requirements.

17 It has been found in communities where electric power is available and generally used, that the consumption amounts to approximately 500 kw-hr. per inhabitant per annum. As stated above, the total output per year of the various power companies connected to the system under discussion now amounts to 2,000,000,000 kw-hr. while the population of the territory served by this system is approximately 11,819,585 showing that there is only 34 per cent saturation in this territory, based on the consumption of electric power in communities where it is in more general use.

MUSCLE SHOALS CONDITIONS

18 Anything that is said or written about the power developments in the Southeast cannot pass without some reference to the Muscle Shoals hydroelectric development, especially that part of this development known as the Wilson Dam or Dam No. 2.

19 The Muscle Shoals section of the Tennessee River is the name applied to the fifty-mile stretch between the railroad bridge

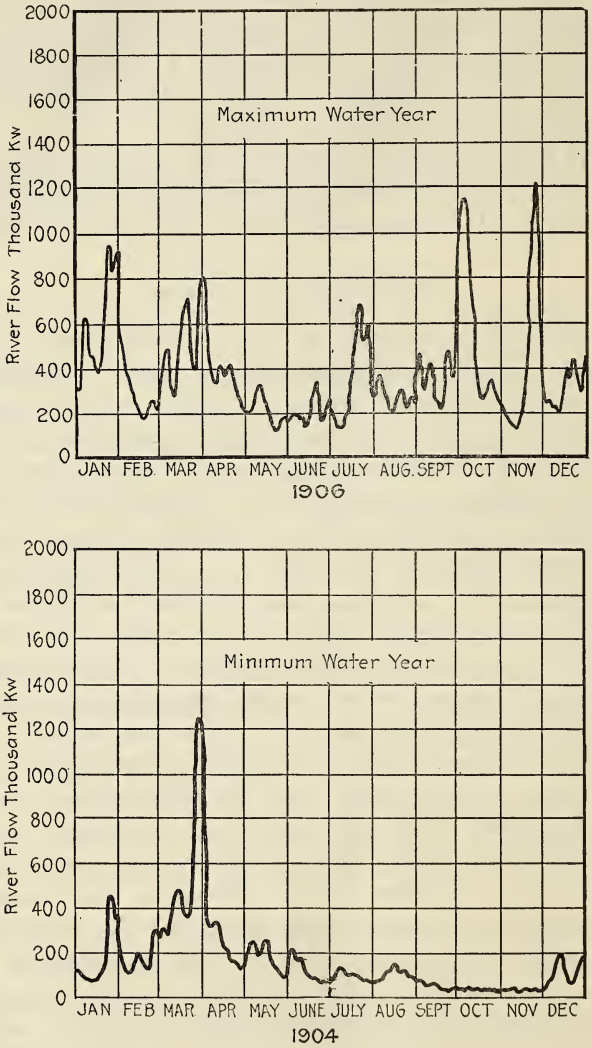

Fig. 3 Hydrographs of the Tennessee River at Florence, Ala., for 1904 and 1906

south of Florence, Ala., and Brown's Island, near Decatur, Ala., which is not now navigable during periods of low water comprising approximately six months of each year. The Shoals are located

approximately 150 miles southwest of Chattanooga, Tenn., and are about 125 miles northwest of Birmingham, and a similar distance

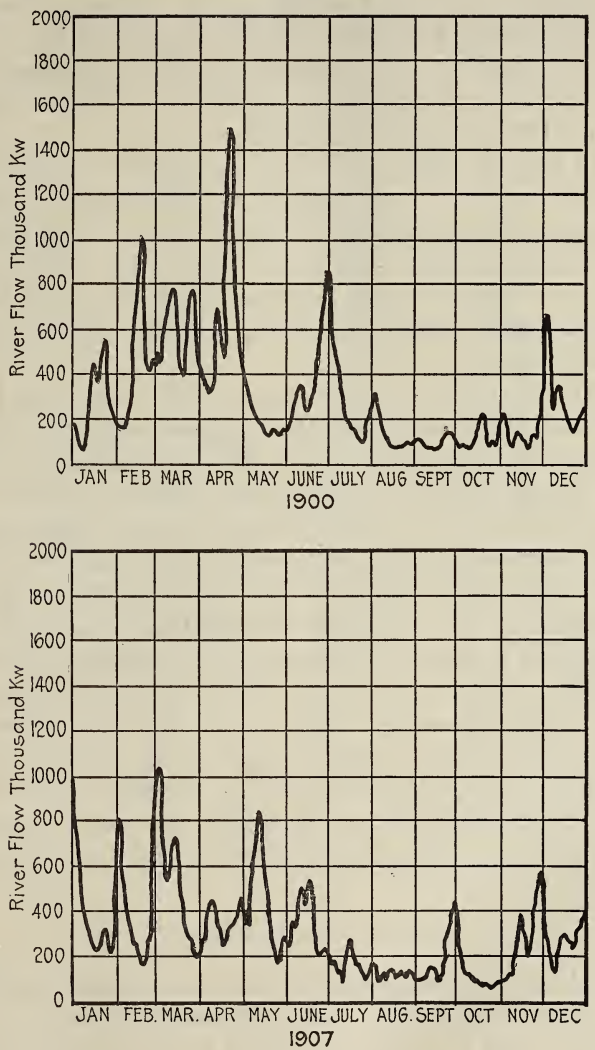

Fig. 4 Hydrographs of the Tennessee River at Florence, Ala., for 1900 and 1907

from Memphis, Tenn. They are about 200 miles in an air line from Atlanta.

20 Sometime prior to 1890 the effort of various individuals

and organizations to do something with Muscle Shoals was begun, and the matter has been more or less active ever since that time. Every effort until recently has been in the nature of forcing the Federal Government to develop Muscle Shoals for the ostensible purpose of improving the navigation of the Tennessee River, on which very little navigation has ever existed, with the secondary object of producing hydroelectric power.

21 In the year 1890 a canal was cut around the Shoals by the Federal Government for the purpose of improving the navigation at this point, and this constituted the first step in the Government's activities at this location. Since that time, various bills have been introduced in Congress for the continued improvement of the river, but all plans have carried with them the idea of developing the hydroelectric power.

22 As soon as it became apparent that the United States would be ultimately drawn into the European war, it was brought to the attention of the Government that an interference with the supply of Chilean nitrates (sodium nitrate, $NaNO_3$) would seriously cripple the ability of the United States to supply ammunition for itself and associated nations in the war. So a clause was inserted in the National Defense Act of 1917, which appropriated twenty million dollars for the purpose of investigating the advisability of, and taking the necessary preliminary steps toward, the erection of a plant for the fixation of atmospheric nitrogen, the idea being that such a plant should be erected at or near some water-power site, remote from the sea coast, where the large amount of electric power required by the process would be cheaply available. The authority for this investigation and action was vested in the President, and he in turn selected a committee to carry out the intention of the Act.

PRESENT DEVELOPMENT

23 An early investigation of the water-power situation at Muscle Shoals indicated that the hydroelectric power plant could not be built in time for service during the war in connection with the nitrate plant, and as a consequence a steam plant was constructed at Nitrate Plant No. 2, with a capacity of 60,000 kw. in one three-part turbo-generator unit, room being provided in the plant for the later installation of a second unit of 30,000 kw. It is a modern, up-to-date plant, well designed and thoroughly efficient. The con-

struction of this plant was carried out by the Government and cost approximately thirteen million dollars.

24 In addition, a steam unit of 30,000 kw. capacity was installed by the Government at the Warrior River steam plant of the Alabama Power Company, 90 miles from Sheffield, Ala., and

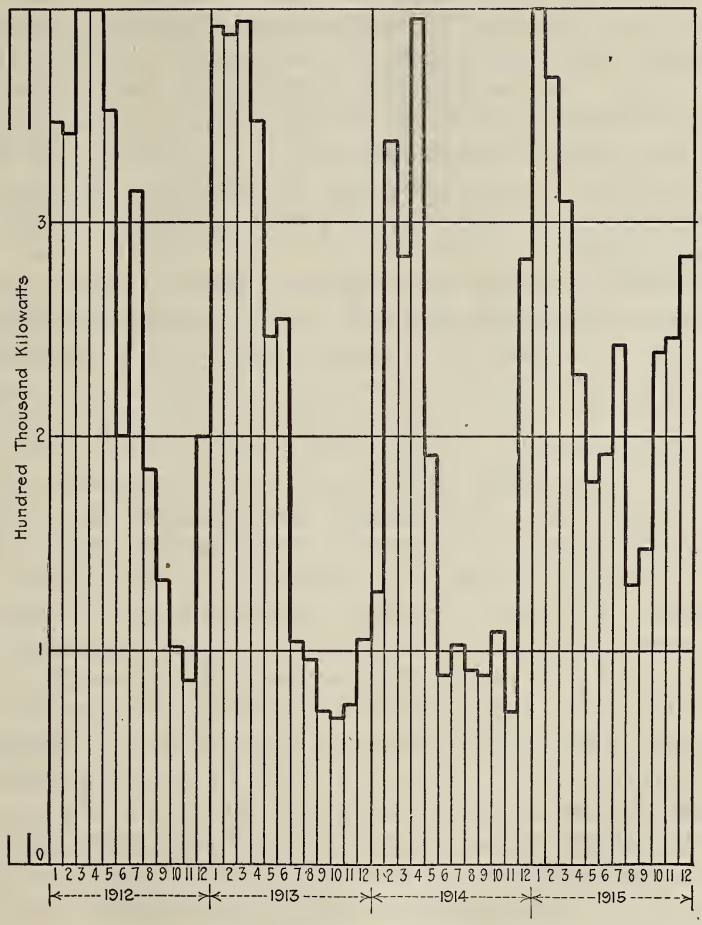

Fig. 5 Powergraph of Tennessee River from 1912 to 1915

connected with the nitrate plant by a 110,000-volt transmission line, the total cost of this improvement being approximately five million dollars. Also, a limestone quarry was purchased and opened at Waco, Ala., for the purpose of supplying the nitrate plant with the lime necessary in the production of calcium carbide.

25 To get back to the subject of the Muscle Shoals hydroelectric development: It was decided early in 1918 to call the dam the Wilson Dam, since the decision as to its construction was originally vested in President Wilson and it was he who directed that the project be carried out. At the time it was decided to proceed with the development, the property was more or less completely in the hands of the Muscle Shoals Hydroelectric Power Company. The company did not desire to stand in the way of any improvement which would further the prosecution of the war, and therefore surrendered all of its holdings, including dam sites, water right and plans to the Government for "one dollar and other considerations." As a consequence of these earlier negotiations, actual construction work on the project was begun during the spring of 1918.

26 The plans call for a dam about 100 ft. high and almost one mile in length, and include two locks for navigation requirements on the northern end of the dam with two 50-ft. lifts, said to be the highest lift locks anywhere now contemplated. The dam is to be fitted with crest gates 18 ft. high for flood control, the gates to be counterweighted and operated by electric motors. The dam is to be of liberal gravity section, and will be constructed with a down-stream apron which extends approximately 100 ft. below the toe of the dam, to protect the river bottom from the overfall. There is to be a highway bridge across the top of the dam above the gates. The dam will contain 1,250,000 cu. yd. of concrete. The power house is to be constructed on the southern end of the structure and built integral with the dam. It will contain a total of eighteen units, four of which have already been contracted for and built. They will be of the usual vertical type with 30,000-hp. water wheels, directly connected to 22,500-kw. generators, together with all the usual auxiliary equipment. It is now contemplated that the fourteen remaining units will be built with 36,000-hp. water wheels, and 27,500-kw. generators, making the total capacity of the plant in generating equipment approximately 400,000 kw. The power house is being constructed at this time for the full ultimate installation.

27 The Government has prosecuted this work somewhat intermittently since it was started, and has completed, up to the present time, a four-track bridge below the dam across the river for construction purposes, and the excavation for the power house. Two

hundred and fifty thousand cubic yards of concrete have been laid in the dam structure on the north end, a portion of the excavation for the locks has been completed, and about 60 per cent of the excavation for the dam, of which there is a total of 620,000 cu. yd.

28 All of the work on this project to date has cost approximately seventeen million dollars, and estimates covering the work necessary to complete the project vary from twenty to thirty-five million dollars. The work today is about 20 per cent complete and has been entirely shut down for more than a year.

29 Much has been said from time to time about the foundations at this dam site. A thorough investigation has been made by both diamond drills and well drills, and the records of these operations indicate that the foundations under the main part of the dam are satisfactory. There is, however, a seam appearing in the southern abutment which extends for a long distance into the surrounding country, and this is now being investigated by tunneling to see if it is or can be made watertight.

POSSIBILITIES

30 Volumes have been written in the Government records and in the press regarding the hydroelectric-power possibilities of Muscle Shoals. It is distinctly a run-of-river proposition, as no storage is created by the dam except that required to control the daily flow of the river. The Government at the present time has a requirement in existence for the benefit of navigation below, which will not allow the control of river flow under 10,000 cu. ft. per sec. The lowest stream flow ever recorded was 6900 sec-ft., which corresponds to a load of 43,125 kw. under the proposed plans for the development.

31 The greatest fluctuation in any stream in the Southeast probably occurs in the Tennessee River. The United States Government in planning a development at Muscle Shoals recognized this fact and contemplated an installation of 400,000 kw., only a portion of which can be depended upon for continuous power. According to the hydrographs of the river at this point, the amount of strictly primary power at Muscle Shoals averages about 100,000 hp., and even this figure is based on the assistance of some steam capacity which must be maintained. It is a source of considerable regret to engineers who are familiar with the Tennessee River to read in

the press of the enormous amount of power allotted to Muscle Shoals development, usually stated as 1,000,000 hp., and the consequent misleading of the general public on this much-discussed project. The development has distinctly a stream-flow characteristic, the discharge of the river varying widely from month to month, and more markedly from year to year. The contemplated capacity of 400,000 kw. at Muscle Shoals could be operated under full load only at widely separated intervals, since the river flow does not equal this amount of power except on an average of less than thirty consecutive days per year.

32 In Fig. 3 are shown hydrographs of the Tennessee River at Florence, Ala., for the year 1906, which is the maximum water year recorded, and for 1904, which is the minimum water year recorded. It will be noted from the curve for 1904 that for a large part of the year the flow at Dam No. 2 would have produced only approximately 50,000 kw., while in the maximum water year, for a large part of the time, the flow falls below 200,000 kw. In studying the hydrographs of this stream covering a period of approximately thirty years, the years 1900 and 1907 have been found to represent about the average conditions of river flow. The hydrographs for these two years are shown in Fig. 4, and it will be noted that the continuous output that can be depended upon throughout these average years is in the neighborhood of 100,000 kw.

33 In Fig. 5 is shown a powergraph of the river for four consecutive years, 1912, 1913, 1914, and 1915, which are typical years based on an installation of 400,000 kw. capacity, as proposed by the United States Government at Dam Site No. 2. It will be noted that in each of these years except one the stream flow at some times falls below 100,000 kw., and that only at widely separated intervals does the stream flow equal or approach the 400,000-kw. mark. It can be seen from these hydrographs that the statement of one million horsepower which has appeared in the press repeatedly, is a gross exaggeration of the output to be expected from the plant at this location. There are many undeveloped water-power sites in the Southeast which can be developed on a commercial basis to much greater advantage than the development at Muscle Shoals.

34 From the foregoing it can be seen that the Southeast is open for great industrial development, as it either has or can develop the necessary hydroelectric power, and enjoys the natural

advantage of an even and salubrious climate, together with a very large store of raw materials in both its agricultural and mineral resources.

DISCUSSION

O. G. Thurlow. Any consideration of the inter-connected system in the Southeast should include Muscle Shoals. The great value of the Muscle Shoals hydroelectric power project comes from its location rather than from its comparative cost of development with other potential powers in the Southeast. It is the largest hydroelectric power resource in the Southeast, and fortunately for those people living in the region to the west and south of the fall line where no hydroelectric power is available, Muscle Shoals lies directly on the fall line. In that region which should be served by Muscle Shoals there has been practically no reduction in the cost of power for the last ten or fifteen years, whereas, throughout the remainder of the southeastern region the cost of power has been greatly reduced due to the development of hydroelectric power. In that region which has been served by hydroelectric power there has been a tremendous advance along industrial lines, whereas, the region which will be the logical market for the Muscle Shoals power has been in a dormant state industrially, although in location, transportation facilities and in resources other than power, this region is more favorably situated than the region to the east which has advanced so rapidly.

It seems reasonable to expect an industrial advance in this region similar to that in the remainder of the Southeast if the Muscle Shoals power is generally distributed throughout the region. Little industrial progress may be expected in west Tennessee, Mississippi and Louisiana if the prime power from Muscle Shoals goes into the manufacture of aluminum or possibly fertilizer. One of the difficult things about the development of any great hydroelectric power is the quick disposal of the available power; one of the chronic difficulties with the development of all the large rivers in the East is the disposal of the secondary or run-of-river power. It so happens that for Muscle Shoals we have a market available for the disposal of the primary power and a chemical plant available for a disposal of a great part of the secondary power.

F. P. Cummings. The paper covers the problem in a comprehensive way. Very little can be added to the general features of the situation but it may be timely, perhaps, to bring out a point in connection with present conditions in greater detail than outlined in this paper.

As stated by the author, the fall of 1921 was marked by a protracted drought throughout the South and particularly on the eastern slopes of the lower Appalachian Mountains. The drought reached such an acute stage early in October that the power companies operating in some of these states were unable to supply their customers from the combined capacities of their steam and hydro plants and many cotton mills and other industries on their lines were closed down and employees engaged in these industries were thrown out of employment.

At that time the power companies in Tennessee, Georgia and the Carolinas were inter-connected and an inter-connecting line between the systems of the Alabama and Georgia companies was being constructed. This line was rushed to completion and pressed into service, resulting in relieving to an appreciable extent the power shortage particularly in the Carolinas from the unabsorbed steam-generating capacity of the Alabama company. However, full service could not be re-established and it was therefore evident that a new source of power would be necessary to protect the service in the various states during the remainder of the dry period of 1921 and the low-water period of 1922, or until hydro developments then under construction could be completed and put in operation.

With the view of relieving the then serious situation and the possibility of providing for conditions that might arise during the succeeding year, steps were taken by the Alabama company to obtain a temporary lease of the 60,000-kw. Government steel plant at Sheffield, Ala. The engineers of the War Department made a careful investigation of the situation which substantiated in every respect the contention of the several power companies as well as many of the larger manufacturing companies which depended upon the power companies for electric service that an actual and serious power shortage existed. The negotiations were finally concluded by the Alabama company taking a revokable lease for the Sheffield steam plant for a period of one year beginning December 1, 1922.

By devoting this Government plant to the industrial activities of the South, it has insured power service to many industries that might otherwise be obliged to curtail their production during the low-water period of this year and it has provided power service for some industrial expansion in the territory covered by the transmission lines connected to this plant.

C. S. Brown said that the Society ought to know something about the great development at Muscle Shoals and it should take some action towards furthering the development of that great project.

It had been his privilege, he said, to be one of the original committee of the Engineering Association of the South, whose report brought about the development of the great water power, as far as it has been brought about, and he felt that an opportunity like the present one should not be lost as the development of Muscle Shoals was a matter of very great moment, not only to this locality, but to this whole country.

It was his opinion that Muscle Shoals had possibilities approaching a million horsepower, which, expressed in terms of coal, meant a waste, through lack of development, equivalent to at least one million pounds of coal an hour. He thought that the Society should appoint a committee of competent engineers to investigate the proposed development and advise Congress concerning it.

George A. Orrok said that the Tennessee River had a mean flow of about 35,000 sec-ft.; a low-water flow of 3000 sec-ft., and a high-water flow of 900,000 sec-ft. The fall available at Muscle Shoals was 70 ft., and at seven horsepower per cubic foot of flow it was easy to figure the horsepower available at Muscle Shoals.

W. F. Wilcox spoke of the difficulties in undertaking water-power developments in the South, and of the progress in the last twenty years. Muscle Shoals, he said, was absolutely necessary to the South and he asked for the coöperation of every engineer, that he give some thought to the industrial development of this section which was handicapped by lack of power. He wished to second Professor Brown's suggestion of a committee of engineers to investigate the project.

Warren M. Wells. The author states that "the first work done on the Muscle Shoals proposition by the Government was in 1890," yet the statement has been published elsewhere that Government records show that the first work was done in 1820 and that General Goethals, then a lieutenant, did additional work there in 1871. In fact the work on Muscle Shoals has been almost continuous since that time although it has always been attacked as a navigation problem instead of a power project.

The author also states that "interested engineers regret that the statement has been made that Wilson Dam No. 2 will furnish one million horsepower as it will furnish only 100,000 primary and 400,000 maximum secondary power." Personally the writer has never heard or seen any statement to the effect that Muscle Shoals Wilson Dam No. 2 would furnish one million horsepower although he has seen the statement that Wilson Dam No. 2 plus Dam No. 3 would furnish 750,000 to 800,000 hp., and that the construction of storage dams further back toward the head waters would retain the flood waters (16 to 20 ft. difference between high and low water at the Shoals) and thus give over 1,000,000 hp.

The Author does not want to give the impression that he is not in favor of the Muscle Shoals development, but he wishes to correct the misinformation on the subject that seems to be prevailing in the minds of people who have not been in touch with it.

Professor Brown talks of a million horsepower. The Government at Muscle Shoals contemplated 400,000 kw. development; and yet on only an average of thirty days a year can that capacity be developed. Muscle Shoals is good for 100,000 kw. or so in primary power, and for much more in secondary power, but no one knows how much more. Secondary power cannot be transmitted to any great extent, because it does not pay to build transmission lines for secondary power. If this secondary power can be used locally, by the Alabama Power Company, if you please, all well and good, but otherwise it cannot be used.

This 100,000 hp., 100,000 kw., rather, needs some assistance to be primary power. There is a 60,000-kw. steam plant that can assist it to that extent.

Mr. Orrok spoke of a minimum flow of 3000 sec-ft. The Government record shows 6900 sec-ft. in 1904 as being the minimum, but on calculation as to what the low flood should have been

there at that time I found it was a little less than that, probably around 5500 sec-ft.

Muscle Shoals is a big power, but it is not to be compared with Niagara, and several other water-power fields in the United States.

No. 1851

NEW DEVELOPMENTS IN HYDROELECTRIC POWER-PLANT DESIGN

By J. A. Sirnit, Birmingham, Ala.
Member of the Society

The most logical and simplest way to maintain the full capacity of a hydraulic power plant during flood periods is to remove the high tail water from the discharge opening. This may be most successfully accomplished by a backwater suppressor utilizing the waste water and the present paper is chiefly devoted to the development and application of this method.

Two testing models are described and the results presented, while the design of the draft-tube orifice is discussed at considerable length. Finally the plant of the Alabama Power Company at Mitchell Dam on the Coosa River in Alabama, where the Thurlow type of backwater suppressor was first conceived and applied, is described, details of its construction and equipment being included.

WITH the rapid development of industry calling for more and cheaper power, and with the increasing cost of steam power, greater interest is being shown in water-power developments all over the world.

2 In this country and abroad hydroelectric developments are being made on a scale never experienced before, and plants are being built and generating units installed in sizes that would have been astounding but a short while ago. Economy in design and efficiency in operation have become factors of great importance, and the designing engineers of hydroelectric power plants are striving to improve the efficiency of the power plant with the same keenness which the designers of the steam-electric power plants have manifested for many years past.

3 Just as the designer of steam plants endeavors to produce the most power out of a pound of coal, so the designer of hydroelectric plants is anxious to utilize every drop of water and con-

Presented at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

vert it into useful power with the minimum loss, and for the least cost.

4 The designers of water wheels and turbines have helped a great deal by improving the efficiency of the driving machinery; the designers of the generators have done likewise and these machines have well-nigh reached the maximum possible point of perfection.

5 A great achievement was attained in recent years by the hydraulic engineer with the improvement of the draft tube, which has helped very materially to increase the efficiency of the power plant. With the introduction of White's hydraucone, Moody's spreading draft tube, and other improved tubes, a distinctive mark has been set in hydroelectric power-plant design, and today we find the hydroelectric plant one of the most efficient power-producing plants.

6 But there are many power plants and possible sites for such plants where the supply of water is very irregular, and where, owing to the nature of the surrounding country, it is impossible to provide sufficient storage for the utilization of all the water flowing in the stream.

7 In many hydraulic power plants, now in operation or proposed, there is, or will be, a great loss of power annually, due to the fact that during flood conditions when water is being wasted over the spillway, the level of the water in the tail race at the outlets of the draft tubes leading from the turbines is raised, thus reducing the head on the turbines and in turn their capacity and the output of the power plant. There are many notable examples of these conditions, and in some cases reduction in effective head on the turbines becomes so great at times as to completely shut down the power plant. The Hales Bar plant on the Tennessee River is an example of where these conditions occur during the spring floods.

8 In many cases of proposed developments, engineers have hesitated in recommending the carrying out of the development on account of backwater conditions in the tail race, which can be foreseen and predicted very closely. The proposed development at the Great Falls on the Potomac, which has been under discussion for many years, represents an example.

9 Engineers for many years have wrestled with this problem and have tried to overcome this difficulty by installing a greater number of generating units where it was possible to do so, to com-

pensate for the loss in capacity of each unit; of course such a procedure necessarily makes the installation more expensive, with the resulting increase of production costs. Attempts have also been made to counteract this loss in head by admitting water into the draft tube through jets at a relatively high velocity which, by accelerating the velocity of the combined turbine discharge and jet water through the draft tube, produce a negative head which is added to the head on the turbine. A number of variations in the place and manner of introducing this jet into the draft tube

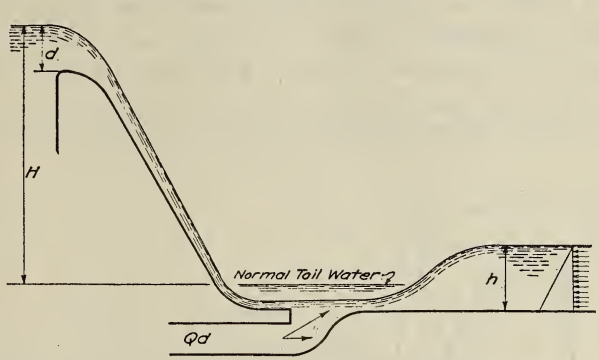

FIG. 1 ACTION OF SPILLWAY WATER ON TAIL WATER

(H = head above tail water; d = depth of water above spillway; h = total water column balanced; Qs = quantity over spillway; Qd = quantity through draft tube; Vs = velocity over spillway; Vd = velocity through draft tube; and w = weight of 1 cu. ft. water = 62.4 lb.)

have been tried and results published. But the results have not been entirely satisfactory.

10 The most logical and simplest way to maintain the normal head on the water wheel, and thereby the full capacity of the plant, during flood periods, is to remove the high tail water from the discharge opening, and credit is due to O. G. Thurlow, chief engineer of the Alabama Power Company, who conceived the idea of utilizing the waste water to accomplish this fact, thereby successfully solving the problem. The development of this conception resulted in what is now known as the Thurlow backwater suppressor.

THE THURLOW BACKWATER SUPPRESSOR

11 It is a well-known phenomenon that water flowing over a masonry dam having a downstream face of ogee section, leaves the apron in a thin sheet at high velocity. At a point below the

dam the water rises turbulently, forming a so-called "standing wave," "hydraulic jump," or "back roll." The thickness of this sheet and the velocity of the water depend upon the height and shape of the downstream face of the dam, upon the depth of water at the crest of the dam, and upon the quantity of water flowing over the dam. The energy developed in this thin sheet of water has generally been regarded heretofore as solely of a destructive nature, but in the backwater suppressor the energy of the spillway water is so directed as to remove the backwater from over the draft-tube orifice, sweeping it downstream, thus freeing the draft tube from this pressure of water over it and maintaining a practically uniform head on the turbine as long as the spillway water is able to sweep the backwater away from the draft-tube orifices. In Fig. 1 is shown diagrammatically the action of the overflow spill-

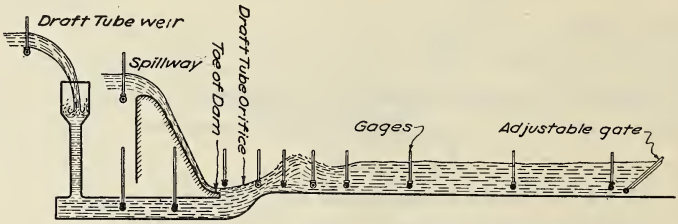

Fig. 2 Design of Testing Flume

way water on the tail water. The energy of the spillway water not only removes the height of tail water from the draft-tube orifice, but even lowers the normal tail-water level to a predetermined depth.

TESTING MODELS

12 In order to substantiate the idea and evolve a definite theory on which to base the calculation, a model was constructed on a 1:24 scale at East Lake in Birmingham, Ala., where a small flow at 3 ft. head was available. The results obtained were so interesting that, to check the data obtained and to increase the accuracy of measurements, a second and larger model was constructed on a 1:10 scale at Jackson Shoals, Ala., where a greater flow and head were available.

13 The results obtained with the two models, when reduced to the same scale, were in close agreement and the observed results followed those calculated for similar conditions.

DESCRIPTION OF MODELS

14 Fig. 2 shows diagrammatically the essential elements of these models. Each consists of a forebay for stilling the water, a spillway having the usual ogee section, a draft tube with its orifice located directly under the spillway, and a tail-water wasteway.

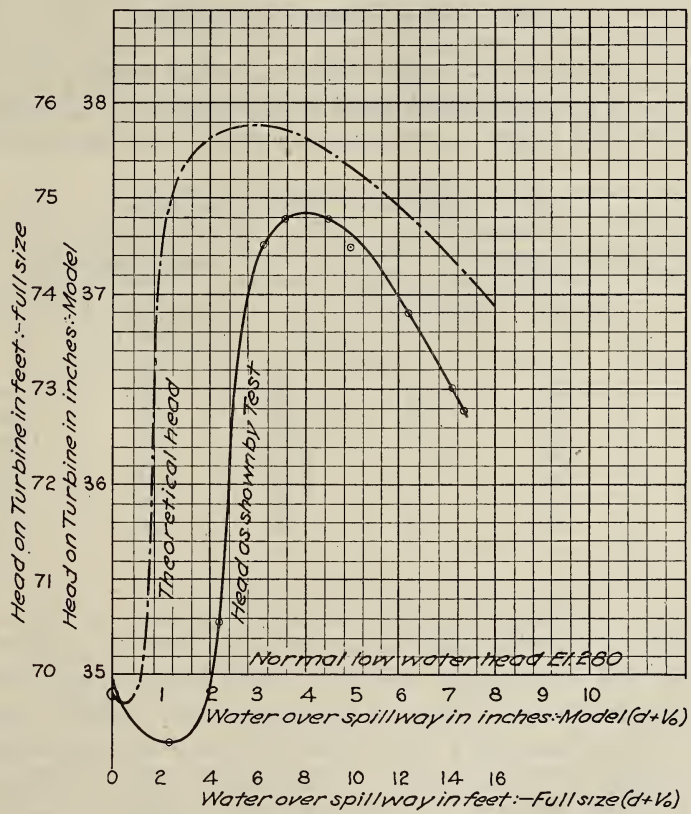

Fig 3 East Lake Head Curve

15 The height of the water in the forebay, wasteway and draft tube were measured in glass gages conveniently located on central platforms. By means of these, measurements corresponding to the pond-level head above the river bed, the effective turbine head, the depth of water over the spillway, and the backwater head could be conveniently determined.

16 Weirs were used in admitting water to the draft tube so

that the amount could be accurately measured. This amount was ordinarily maintained to correspond to a normal turbine discharge.

17 Bear-trap stop logs were provided at the end of the wasteway by which the height of the water in the wasteway could be varied to simulate backwater.

CALCULATED AND OBSERVED RESULTS

18 The height of the standing wave, taking a section of unit width (1 ft.) and neglecting losses due to the friction of the moving water over the spillway, can be approximated by the empirical

TABLE 1 RESULTS OF TESTS ON MODELS

East Lake Model			Jackson Shoals Model		
d, in.	Observed h, in.	Calculated h, in.	d, ft.	Observed h, Corrected ft.	Calculated h, ft.
4	6.00	6.5	0.25	0.40	0.59
5	7.75	8.05	0.50	0.84	0.88
6	8.75	8.9	0.75	1.30	1.20
7	9.75	10.0	1.00	1.51	1.56
7.5	11.00	10.6	1.25	1.72	1.78
....	....	....	1.50	2.10	2.08
....	....	....	1.75	2.48	2.35

formula $h = \sqrt{d^{\frac{3}{2}}H^{\frac{1}{2}}}$, in which d is depth of water in feet over the spillway and H the head in feet above the spillway apron. This does not take into account the effect of the turbine water. This factor will increase the height of standing wave, so that the actual results will be somewhat better than as shown by the formula. A comparison of the calculated and observed heads of backwater, which different amounts over the spillway are capable of removing from the draft-tube orifice, is given in Table 1. These figures are for a section of backwater 20 per cent greater in width than the spillway and are calculated from the formula $h = \sqrt{0.83d^{\frac{3}{2}}H^{\frac{1}{2}}}$.

19 The figures of Table 2 were compiled from three separate tests made at different times and show the performance of the Jackson Shoals model with 18 in. of water over the spillway in suppressing various heads of backwater.

20 The effect on the turbine head of different amounts of

water over the spillway of the East Lake model is shown in Fig. 3. It should be noted that very small amounts cause a decrease in head because sufficient velocity has not been given to the draft-tube water to cause a thinning of the sheet over the draft-tube orifice, but when the ratio of spillway to draft-tube water increases to about one to one the curve takes a decided course upward and reaches its peak and starts falling off again due to a greater quantity of water passing over the orifice, with practically no increase in velocity above that of the point of maximum head. This curve does not represent the best performance possible but it is typical of the ideal theoretical curve which is shown in the same figure.

21 The curve in Fig. 4 shows the overall performance of

TABLE 2 SUPPRESSING PERFORMANCE OF JACKSON SHOALS MODEL

Water over river bed, in.	Head without suppressor, in.	Head with suppressor, in.	Water over river bed, in.	Head without suppressor, in.	Head with suppressor, in
0	96	....	24¾	71¼	88
½	95½	96½	26	70	87
2¾	95¼	95¾	27½	68½	80
19	77	96¾	29	67	73
20¼	75¾	96½	29¼	66¾	74
21½	74½	97	30¼	65¾	73
22	74	96½	34	62	68¼
22¼	73¾	96¾	35¼	60¾	66
22¾	73¼	96¾	40½	55½	66¾
23¾	72¼	96½	46	50	56½
24½	71¾	89½	....	....	....

the test as compared to that without the suppressor. It will be noticed that the head drops off slightly as the backwater increases, due to a greater quantity of water necessary to hold the standing wave, until a point is reached where the standing wave is 13.26 in. high, and the curve turns sharply down. This point represents the critical stage or the maximum height of backwater capable of being suppressed by a given amount of spillway water. This could be increased by having a design of spillway permitting a greater discharge.

22 Even after this point is reached a substantial increase in head is obtained, and at the last point plotted, with backwater 50 per cent of the total head, an increase of 5 in. is recorded. Fig. 5 illustrates the condition of backwater after the critical stage has been passed.

DESIGN OF DRAFT-TUBE ORIFICE

23 Several shapes of draft-tube orifices were tested. That shown in Fig. 6 leaves very little to be improved upon for getting a maximum height of standing wave and a maximum head on turbine. This design can be modified to give a greater head on the turbine but only at a sacrifice in the height of the standing wave.

24 The location of point x is of great importance as it controls the increase in head on turbine, and its relation to the elevation and angle of discharge of the spillway apron must be very

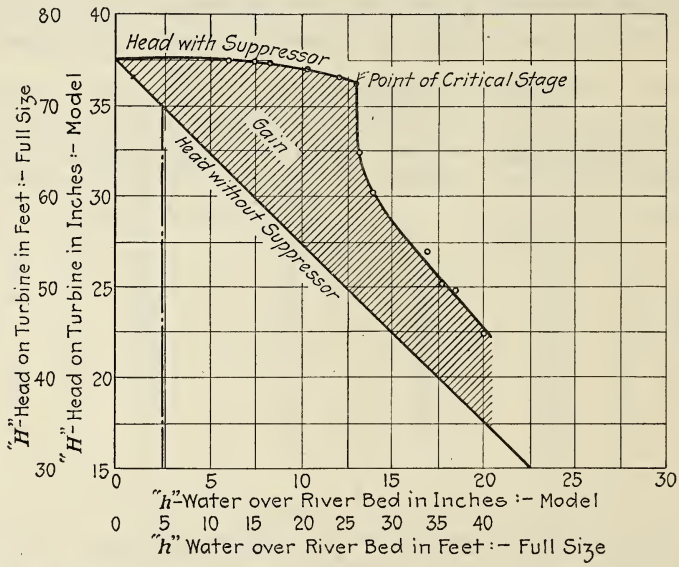

Fig. 4 East Lake Performance Curve

exact or a poor performance will result; also consideration for the turbine "run-off" during the period when the suppressor is not in action must be kept in mind.

25 The length of tangent t must be such that it will direct the spillway water well along the line ab, and it should be at least 5 ft. for heads up to 25 ft. and 7 ft. to 10 ft. for heads varying from 30 to 100 ft. The amount of water flowing over the spillway will cause this length to change, but for depths up to 15 ft. this range will hold good.

26 The distance out of point x is determined by the amount of water passing through the draft tube. The area of the draft-

tube orifice subject to the action of the sheet of spillway water must be proportioned so that a back pressure in the draft tube is not necessary to force the two waters to mix. The curve of R_3 is great enough so that the centrifugal force of the fast-moving sheet of water will not cause it to leap clear of the concrete surface. R_2 makes a smooth transition, connecting point x and R_3 and is about $\frac{1}{2}$ R_3.

27 The standing wave is held out just beyond tangent R_3. The total distance from the lip of spillway apron to the point of wave must not be greater than good design of the other features will permit, as the friction of the high-velocity water diminishes

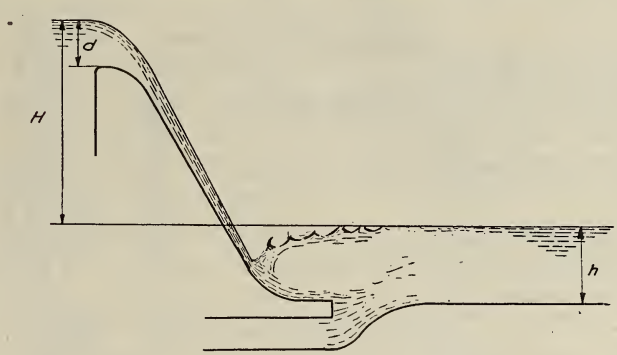

Fig. 5 Flood Conditions of Model

the height of the standing wave, thereby causing a greater amount of waste water necessary to operate.

28 The river channel below tangent R_3 must be finished with neat concrete for a distance of four heights of the standing wave, in the case of heads up to 25 ft., and five to six heights for heads varying from 30 to 100 ft. This is necessary for good performance although a rough bed or water pocket can be used, but a great waste of water is required and heavy undercurrents, due to an imperfect standing wave, might scour the river bed badly or undermine the protecting apron.

29 This is of great importance if the rock strata are poor, as open seams will allow the high static pressure beyond the wave to be transmitted back under the protecting apron causing an uplift. This uplift must be given attention even when the apron is carried out to the desired point and provision made for its secure anchor.

DIVIDING WALLS

30 Each unit must be provided with dividing walls to prevent the water from coming in on the sides and to allow any one unit to operate independently.

31 The correct height and length of these walls is determined by the backwater and discharge curves of the river at the location of the power house. Section *A-A*, Fig. 6, shows the condition for the installation of four units. The three intermediate walls are at an elevation shown by the backwater curve when the required amount of water to operate four turbines and three sup-

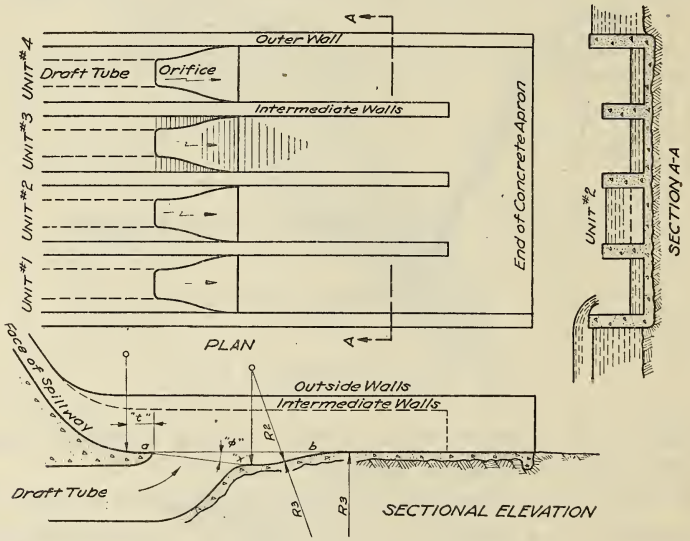

Fig. 6 An Excellent Design of Draft-Tube Orifice

pressors is passing that point. When the fourth suppressor is brought into action the condition is as shown by the dotted line in unit No. 2, and these walls simply act as a guide to hold better conditions at the point of wave and prevent surging.

32 The two outer walls are designed for extreme flood conditions, as any water passing over the walls into the suppressor sheet will cause a loss in head on the turbine and in height of standing wave. However, if there should be spillway gates on both sides of the power-house section equal to the width of two units, it need not be at the maximum high-water elevation but at the

stage shown by the combined discharge passing the power house and side spillway. This is due to the fact that although a perfect standing wave is not formed at the toe of the side spillways, a lower elevation is created than will be further down stream, where the true backwater elevation is measured.

POWER PLANT WITH BACKWATER SUPPRESSOR

33 Encouraged by the splendid results of the model tests and the economic advantages of the undertaking disclosed by careful analysis, the Alabama Power Company decided to apply the backwater suppressor at Mitchell Dam on the Coosa River in Alabama, which is now under construction.

34 The estimates showed that not only was the cost of the suppressor-type plant not higher than that of a conventional-type plant of equal output, but even lower in cost for the same number of units, and therefore for a given output the suppressor-type plant was considerably less expensive.

35 The design finally evolved for the above-mentioned power plant with the backwater suppressor shows several radical changes and represents an entire departure from the established conventional design of hydroelectric power plants.

36 The principal and most outstanding feature in this plant is the location of the power units on separate foundations in the river on the upstream side of the dam (Fig. 7). Although this was not necessarily the only possible way to build the power house, it was the most economical. The river at this point being comparatively narrow, the entire length of the dam was required for spillway, and since the prime requisite of this type of plant is that the draft tube discharge directly under the spillway section, the units had to be so located that they would not obstruct any part of the spillway section.

37 The individual and separate power-house units offer the added advantage that trash racks do not present a solid front, which helps materially in cleaning them and affords an easy and convenient way of diverting the trash past the power house and over the spillway.

38 The usual power-house building is entirely eliminated. The generator room is covered with a low roof, which is designed in two sections, joined on the transverse line, mounted on rollers; each section moving in opposite direction. In normal operation the

generator room is completely protected from the weather, but for handling large parts of the machinery, this roof can be opened and a crane utilized for performing the necessary work.

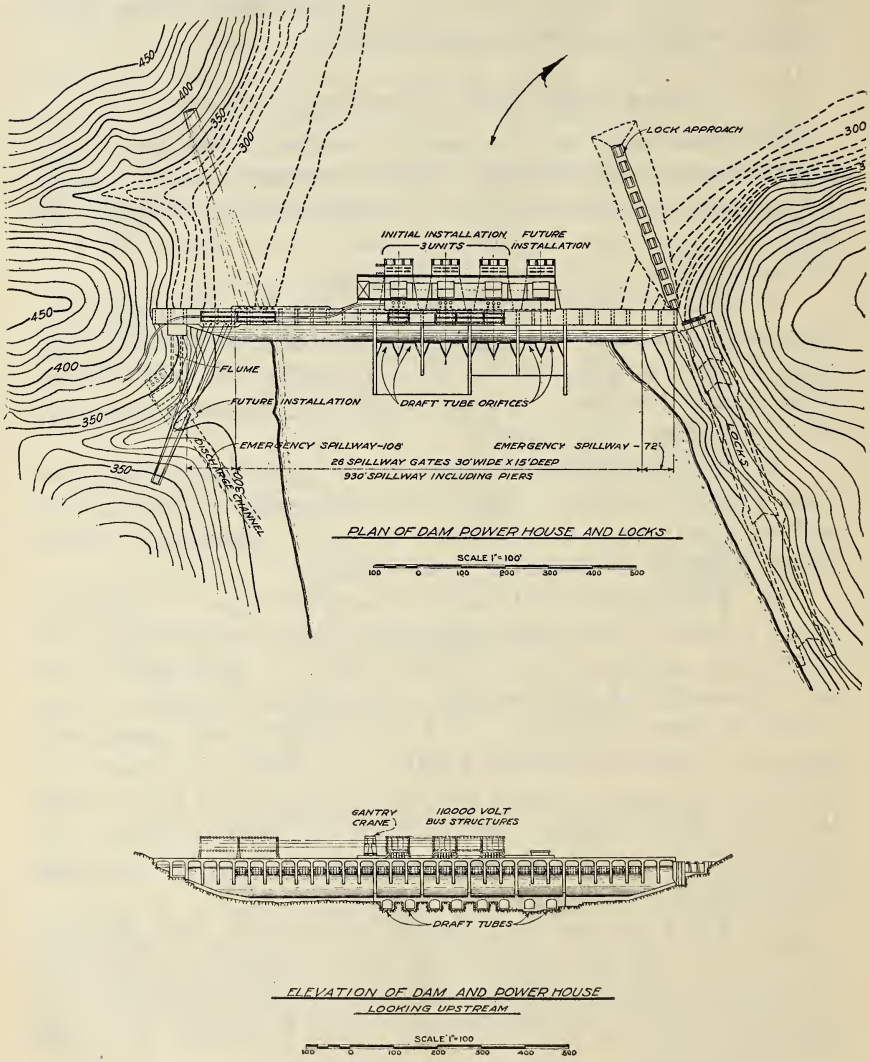

Fig. 7 General Plan and Elevation of Dam and Power House

39 A single gantry crane (Fig. 7) with full travel over the entire length of power-house section is so designed that everything, with exception of spillway gates, can be handled by this crane. It

is equipped with one 125-ton and one 20-ton hook, the former for handling heavy machinery and the latter for raising and lowering penstock gates, handling racks, stop logs and all other lighter parts.

40 An outdoor transformer and high-voltage switching station is located on the bridge over the spillway (Fig. 8). This plant is designed for operation on unit system, i.e., the generator and its bank of step-up transformers are connected as a unit and all principal switching is to be done with the high-voltage circuit-breakers. In stations where units are of large capacity, as in the plant under discussion, (20,000 kva.) the unit system is very economical and preferable, as it presents the simplest, yet sufficiently flexible, form of operation.

41 The entire operating floor is on one floor level. The generators, governors, switchboard, low-tension switches, bus galleries and offices are easily reached without climbing stairs or ladders. This materially adds to the convenience of operators and makes the supervision and inspection of the plant more effective. This feature of providing easy accessibility to all important parts of the station is very frequently lost sight of by power-house designers and the lack of these conveniences has been the cause of inefficient operation or even of breakdowns, due to the fact that operators have neglected the equipment to a lesser or greater degree.

42 The generators are so installed that the hot air from the generator is discharged into a separate compartment under the main floor and is expelled into the atmosphere through side openings of that room in summer time, thereby keeping the generator room reasonably cool. In winter, however, when heat in the rooms is desired, the outside openings can be closed and through registers in the floor the warm air enters the generator room.

43 The penstock does not represent a true scroll casing but rather a combination of scroll effect and open flume. The water velocities in the penstock being low, approximately 4 ft. per sec., this combination affords a better design.

44 The location of power-house units on the upstream face of the dam lengthened the discharge tunnel to about 120 ft. in length from the center line of the unit. Although the velocity of the discharge is only 4 feet per sec., the effect of this long column of moving water had to be carefully investigated and studied. This problem was solved satisfactorily by increasing the governor

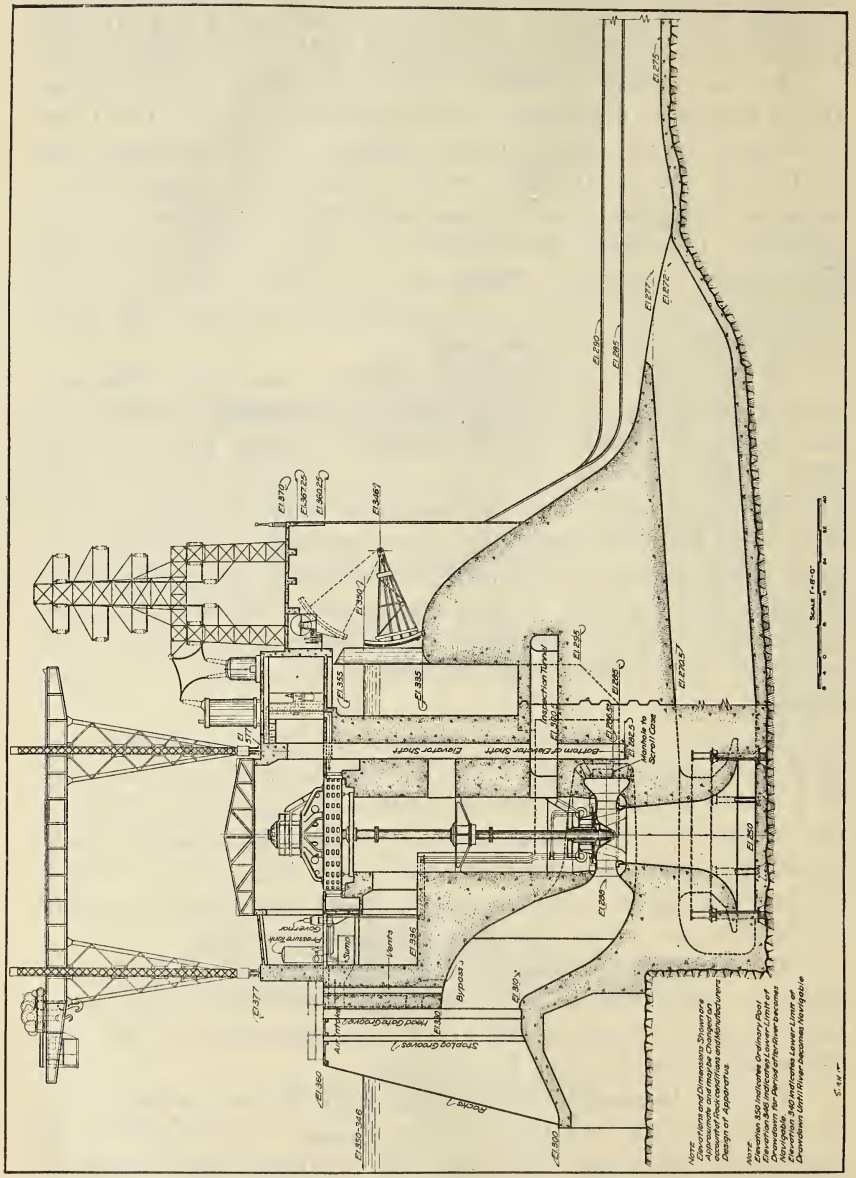

Fig. 8 Cross-Sectional Elevation View of the Power House

action to $3\frac{1}{2}$ sec. and setting the turbine to the lowest practical level. Turbine settings always should be as low as possible, especially where improved draft tubes are employed. Many of the troubles in existing power plants, such as pitting of runners, excessive vibration, etc., can be directly traced to high wheel settings, particularly with high-specific-speed runners.

45 Fig. 7 shows that 26 spillway gates of the Tainter type are provided, each 15 ft. high and 30 ft. wide. These gates are capable of passing the maximum flood water known to have existed in the past. There are also six bays, two on the east and four on the west end of the dam, which are designated as emergency spillway openings. The crest of these openings is level with the top of the spillway gates and their function is to provide additional spillway capacity in case of unprecedented floods when water would rise above normal pool level.

46 Although little has been said about the electrical equipment, it is one of the most important parts of the installation and should always receive close study and serious consideration if economical and efficient operation is expected.

No. 1852

REPORT UPON EFFICIENCY TESTS OF A 60,000-KW. CROSS-COMPOUND TRIPLE-CYLINDER STEAM TURBINE

By Herbert B. Reynolds and Walter F. Hovey, New York, N. Y.
Associate-Members

The present paper contains the results of tests on the 60,000-kw. Westinghouse turbine installed at the 74th Street power station of the Interborough Rapid Transit Company, New York City.

This turbine is of the cross-compound, triple-cylinder type, consisting of one high-pressure element and two low-pressure elements, each element driving a separate generator. The normal steam pressure at the throttle is 220 lb. abs. with a superheat of 150 deg. fahr. exhausting into a vacuum of 29 in. Hg. The speed of all three elements is 1,500 r.p.m.

The primary steam inlet takes care of all loads up to 40,000-kw. A secondary inlet which bypasses the first six stages cares for all loads up to 50,000-kw., after which a tertiary inlet which bypasses the first ten stages opens and increases the capacity to 60,000-kw. The generators are able to carry a maximum load of 70,000-kw. for two hours, this being effected through a hand-operated quaternary valve which bypasses four additional stages of the high-pressure turbine.

The lowest water rate obtained while operating the complete unit under normal conditions was 11.00 lb. per kw-hr., while the highest Rankine and thermal efficiencies obtained were 76.0 per cent and 25.1 per cent., respectively. With the high-pressure and one low-pressure turbine in service the lowest water rate was 11.25 lb. per kw-hr.

Results of tests of 30,000-kw. Westinghouse cross-compound turbines at the same station were presented before the Society in 1916 in a paper by H. G. Stott and W. S. Finlay, Jr., and of 30,000-kw. General Electric turbines of the straight Curtis impulse type in a paper read at the Spring Meeting, Chicago, May, 1921, by the senior author of the present paper.

A PAPER was presented before the Society [1] by H. G. Stott and W. S. Finlay, Jr., in 1916, which reported the results of a series of tests on a 30,000-kw. Westinghouse cross-compound turbine. This turbine was one of three which were installed at the 74th Street Power Station in order to provide additional power capacity for

[1] Trans. Am. Soc. M. E. vol. 38, page 655.

Presented before the Metropolitan Section of the A.S.M.E., New York, February 24, 1922, and at the Spring Meeting, Atlanta, Ga., May 8 to 11, 1922, of The American Society of Mechanical Engineers.

the new subways constructed in New York City during the period from 1913 to 1921 and operated by the Interborough Rapid Transit Company. Soon after the completion of these three turbines work was started on the installation of three 30,000-kw. General Electric turbines at the 59th Street Power Station, while the installation of a 60,000-kw. Westinghouse cross-compound triple-cylinder turbine

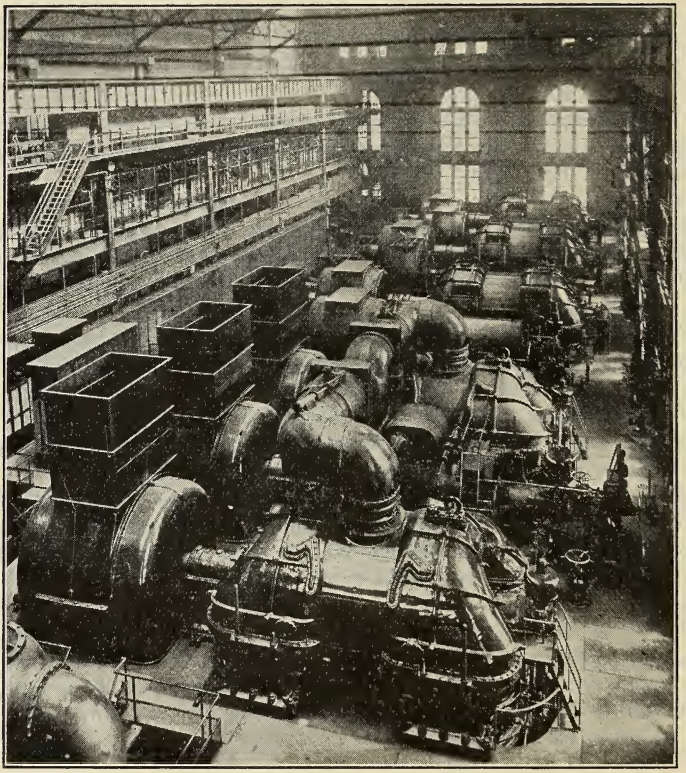

Fig. 1 General View of Turbine

was started at the 74th Street Power Station. A paper was presented before the Society at the Spring Meeting in 1921 which reported the results of the tests on one of the General Electric turbines mentioned above, while the present paper contains the results of the tests on the 60,000-kw. Westinghouse unit installed at the 74th Street Power Station.

2 The engine room at the 74th Street Power Station originally

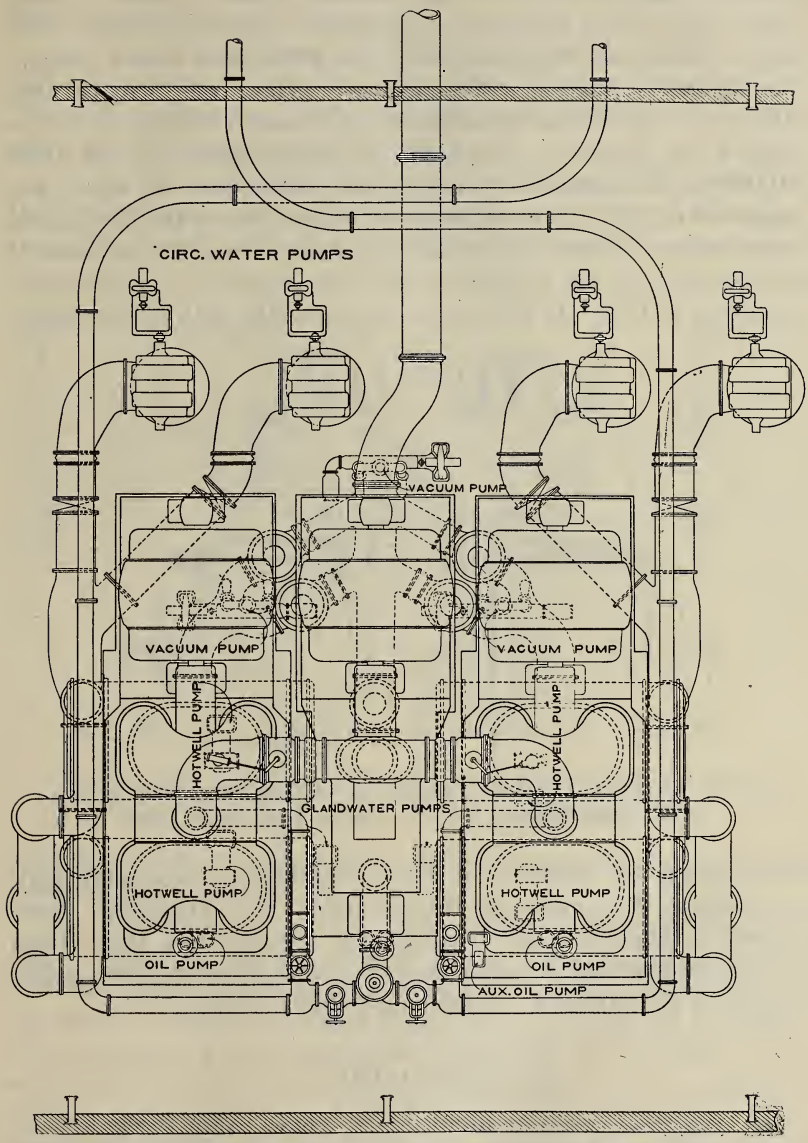

Fig. 2 Plan View of Turbine and Auxiliaries

contained eight 7500-kw. Allis-Chalmers double angle-compound engine units of the well-known Manhattan type and similar to the engines which were later installed in the 59th Street Power Station. During 1905 a 5500-kw. Westinghouse turbine was installed, the capacity of which was subsequently increased to 8000-kw. by rewinding the generator. In order to provide space for the three 30,000-kw. Westinghouse turbines, the installation of which was completed in 1915, it was necessary to remove four of the Manhattan engine units. A fifth engine unit was removed in 1916 in order to provide space for the 60,000-kw. unit, the installation of which was completed in 1918. It will thus be seen that the 74th Street engine-

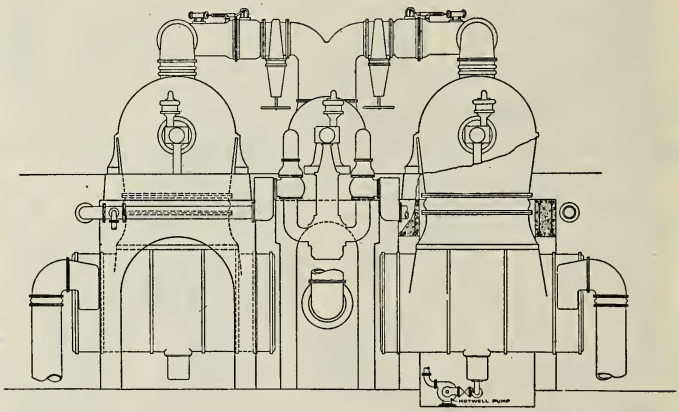

FIG. 3 END ELEVATION OF TURBINE AND CONDENSERS WITH RIGHT-HAND SIDE BROKEN AWAY TO SHOW EXPANSION JOINT

room equipment consists of three 30,000-kw. cross-compound-two-cylinder, Westinghouse turbine units, one 60,000-kw. cross-compound, three-cylinder Westinghouse turbine unit, three 7500, kw. Manhattan type Allis-Chalmers double angle-compound engine units, and one 8000-kw. Westinghouse turbine unit.

THE TURBINE UNIT

3 A general view of the turbine is shown in the foreground of Fig. 1. The three 30,000-kw. cross-compound turbines mentioned above are shown in the background. The 60,000-kw. turbine is of the cross-compound, triple-cylinder type, consisting of one high-pressure element and two low-pressure elements, each element driving a separate generator. Fig. 2 shows the general arrangement

of the turbine together with the auxiliaries, while Figs. 3 and 4 show the front and side views.

4 The normal steam pressure at the throttle is 220 lb. abs. with a superheat of 150 deg. fahr., exhausting into a vacuum of 29 in. Hg. referred to a 30-in. barometer at 58.1 deg. fahr. The speed of all three elements is 1,500 r.p.m.

5 In Fig. 2 it will be seen that steam is supplied to the unit through two steam lines each one of which is provided with a throttle. After passing through the throttles the steam enters a common

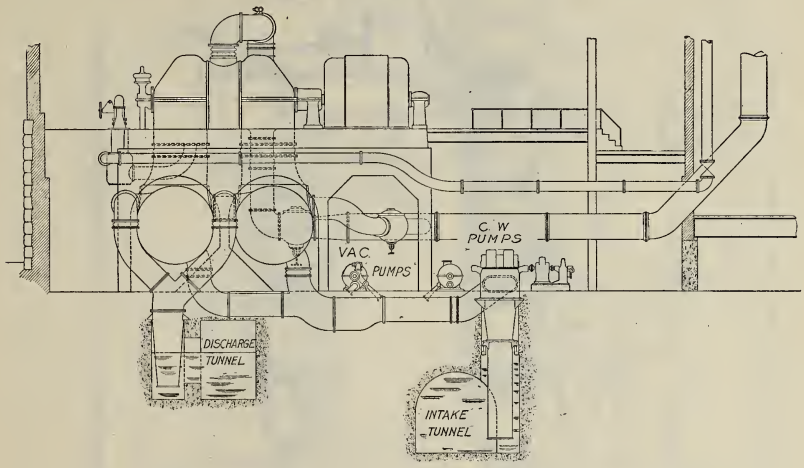

Fig. 4 Side Elevation of Turbine and Condensers Showing Circulating-Water and Atmospheric Exhaust Piping

chamber. The use of two small lines gives greater flexibility and increases the reliability somewhat.

6 A cross-section of the high pressure element is shown in Fig. 5. It will be seen that this element is of the single-flow straight reaction type having a total of twenty-five stages. In addition to the primary steam inlet which takes care of all loads up to 40,000-kw., secondary and tertiary inlet connections are provided. The secondary inlet bypasses the first six stages and is opened automatically by the governor at a load of approximately 40,000-kw. This secondary inlet takes care of all loads up to 50,000-kw., after which the tertiary inlet, which bypasses the first ten stages, opens and increases the capacity to 60,000-kw. The generators are able to carry a maximum load of 70,000 kw. for two hours. In order to carry this load, a hand-operated quaternary valve has been provided

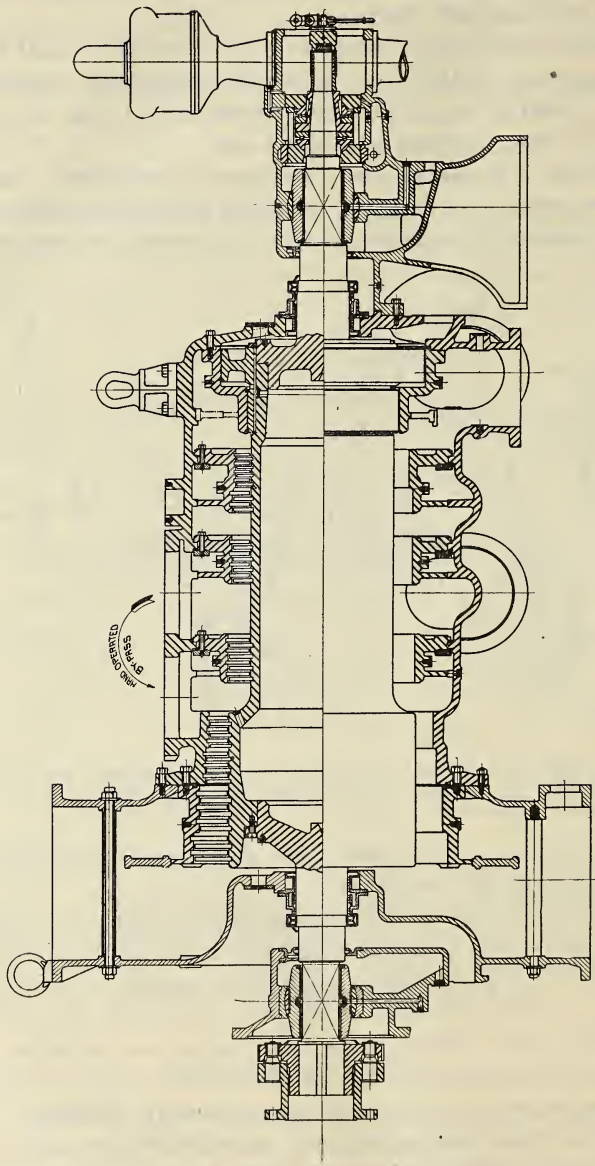

Fig. 5 Cross-Section of High-Pressure Turbine

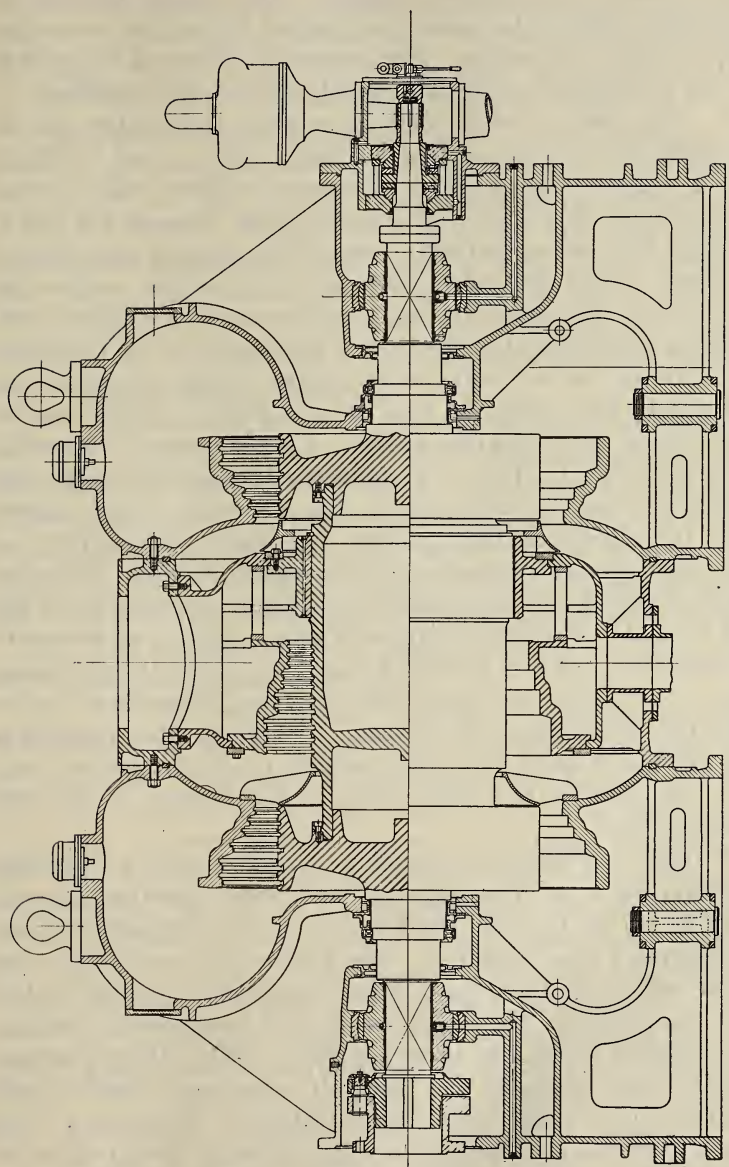

Fig. 6 Cross-Section of Low-Pressure Turbine

which bypasses four additional stages. After passing through the high-pressure element the steam is exhausted through the connection shown on top of the casing. The connection shown at the bottom is for an atmospheric relief valve which has not been installed.

7 The steam is led to each low-pressure turbine through the large pipes shown on top of the turbines in Figs. 1 and 3. Fig. 6 shows a cross-section of the low-pressure turbines, which are of the semi-double flow type. The steam enters through the top of the casing and passes through a single-flow element consisting of ten stages. After passing through this intermediate section the steam divides and passes through two low-pressure sections, each containing six stages, after which it is discharged to the condenser.

8 All glands are water-sealed during regular operation, the water supply being obtained from the condensate. While starting steam is used for sealing the glands. A self-contained lubricating system is provided. The oil is circulated through the coolers and bearings by means of three pumps, one connected to each turbine shaft. In addition to these pumps, two duplex steam-driven oil pumps are provided for starting and emergency service. These two emergency pumps are controlled automatically by floats in an elevated oil tank. The unit is provided with a number of automatic features which makes it possible for any one of the three elements to go out of service and allow the remaining elements to remain on the line to take care of the load. It has not been possible to make full use of these automatic features up to the present time, due to the absence of a relief valve on the exhaust of the high-pressure turbine.

9 Each low-pressure element is provided with a live-steam connection and a governor adjusted for a wide regulation, approximately 12 per cent., while the governor on the high-pressure unit is adjusted to a close regulation. The governor on each low-pressure element will in the outer part of its travel close a butterfly valve in the line between the high-pressure and low-pressure turbines. These butterfly valves may be seen in Fig. 3. When these governors reach the inner positions they begin to admit and regulate high-pressure steam to the low-pressure turbines. There is a zone between these two positions in which the motion of the governor has no effect. This zone is sufficiently wide to cover the total speed variation of the machine when operating normally so that the low-pressure governors have no function to perform during normal operation.

10 In order that the switchboard operator may keep the low-pressure governors in the middle of the neutral zone, a remote indicating system is provided which gives an indication on the switchboard of the exact position of these governors.

11 Each element is provided with a stop governor which closes the automatic throttle admitting live steam to the low-pressure turbine and the butterfly valve admitting low-pressure steam to the low-pressure turbines only, while the automatic stop governor on the high-pressure will close the main automatic throttle valve only. There is a switch arranged in connection with each of the three automatic throttle valves so that when the automatic throttle valve ope ates, this switch opens the oil switch of whichever turbine element is involved.

12 As mentioned before, the exhaust of the high-pressure turbine is to be provided with a relief valve which, in the event of one of the low-pressure turbines being thrown out of service by the closing of the butterfly valve, will permit the surplus exhaust steam to escape to the atmosphere. This valve will be set to that pressure corresponding to the maximum load on one of the low-pressure turbines which the generator can sustain without injury for a brief period.

13 The automatic features described above will take care of the following emergency conditions:

a If the oil switch to one low-pressure generator is opened, that element will speed up until the low-pressure governor comes into action and closes the butterfly valve in the low-pressure steam inlet. The turbine will then slow down until the governor travels to its other extreme position when the live-steam connection will open and keep the turbine running until shut down by hand. In the meantime the load on the high-pressure and remaining low-pressure turbine will be increased due to the loss of one low-pressure. All of the steam from the high-pressure turbine will endeavor to pass through the remaining low-pressure, which may be overloaded in the absence of the relief valve, which will limit the steam pressure and thus the load which may be applied to the low-pressure turbine. The capacity of the high-pressure and one-low pressure, when operating in this manner, is approximately 35,000-kw.

b If one of the low-pressure turbines speeds up and knocks out the safety trip the butterfly valve will close, the live-steam connection will be prevented from opening, and the oil switch to the element involved will be opened, all of which will permit the element to come to rest.

c If the oil switch to the high-pressure element is opened this element will tend to speed up which will cause the high-pressure governor to reduce the supply of steam passing through the throttle to a very small amount. This will cause the low-pressure turbines to slow down, due to the load which they are carrying, until their governors come into action and admit live steam. The speed may then be brought back to normal by adjusting the low-pressure governors.

d If the automatic stop governor on the high-pressure turbine is knocked out due to overspeed, the oil switch to this element will open allowing it to come to rest. In the meantime the low-pressure governors will come into action and open the high-pressure steam connections to the low-pressure turbines so that they will continue to carry load, the capacity of each low-pressure turbine being approximately 23,500-kw.

14 Pending the installation of the relief valve, it is impossible to take advantage of these various features. In order to prevent any damage which may result from the closing down of any one element, the overspeed trips are tied in together so that if any one element knocks out its stop governor, the entire unit will be shut down.

15 One of the low-pressure elements is provided with a heat balance connection through which any excess auxiliary steam may be injected.

16 The generators are three-phase star-connected, generating 25-cycle current at 11,000 volts. Excitation current is furnished at 250 volts. The generators are cooled by a circulation of air maintained by a fan which forms an integral part of the generator. The air is drawn from the turbine-room basement and discharged from the top of the generator into the turbine-room through a short stack which may be seen in Fig. 1. Each generator is connected through an oil switch to a short bus which bus in turn is connected to the station bus through another oil switch. This arrangement permits all three generators to be brought up to speed and synchronized simultaneously.

CONDENSERS AND AUXILIARIES

17 The condensing equipment for each low-pressure turbine consists of a two-shell, two-pass Westinghouse condenser, two Westinghouse centrifugal circulating pumps each driven through reduction gears by Westinghouse turbines, and two Westinghouse

centrifugal condensate pumps each driven by a Westinghouse turbine. The entire unit is served by three LeBlanc air pumps.

18 Each condenser shell contains 25,000 sq. ft. of tube surface made up of 5,300 tubes, 18 ft. long, 1 in. outside diameter and of No. 18 B.W.G. thickness. A portion of the tube surface next to the turbine exhaust amounting to 1,200 sq. ft. is used as a

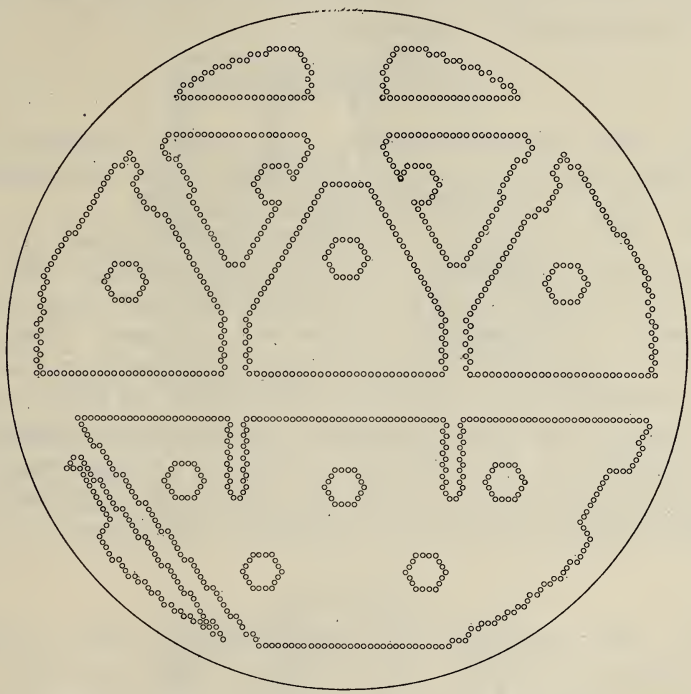

Fig. 7 Diagram Showing Tube Location in Condensed Head

preheater through which the condensate is circulated. Fig. 7 shows the arrangement of the tubes.

19 Each condenser is provided with two relief valves as shown in Fig. 2. The reason for using two small valves instead of one large valve is that these valves were available from the old engine units.

20 Each circulating-water pump is capable of delivering 28,000 gal. of water against a total head of 20 ft. The turbines which drive these pumps operate at 3,600 r.p.m., which speed is reduced to 360 r.p.m. through Westinghouse gears. Each condensate pump is capable of delivering approximately 1000 gal. per

min. against a discharge head of 40 ft. The turbine and pump both operate at a speed of 1,600 r.p.m.

21 The air pumps are of the well-known LeBlanc type operating at 2,500 r.p.m. They receive their hurling water from the discharge of the circulating-water pumps. In order that the air leakage may be checked from time to time, connections have been made to a rotative dry vacuum pump, belonging to an adjacent unit, which can be discharged into a gasometer.

FOUNDATIONS

22 Each element is mounted on a separate structural steel foundation encased in concrete. The steel of these foundations is shown in Figs. 8 and 9.

23 The condensers are hung from these foundations by steel straps bolted to pads on the upper part of the condenser shells. In view of this rigid mounting it was necessary to install a large rubber expansion joint between the turbines and condensers. This joint is shown in Fig. 10. It will be noticed that water protection has been provided so as to protect the rubber from the high temperature during the starting up periods. Rubber expansion joints are also provided in the circulating water-lines. Copper expansion joints are inserted in the steam lines between the high and low-pressure turbines and also in the atmospheric relief lines.

TURBINE TESTS

24 Although the turbine was first placed in complete operation on October 11, 1918, the tests were not run until July 1921. The equipment used for conducting the turbine tests consisted of two large water tanks for measuring the steam consumption, three single-phase rotating standard watthour meters for measuring the output, and all the necessary thermometers, gages and mercury columns for determining temperatures, pressures and vacua as recorded in this report.

25 The water-measuring tanks had a capacity of 300,000 lb. each. These tanks were very carefully calibrated before the tests by filling them with water weighed on platform scales. The condensate was measured throughout the tests, by alternating from one tank to the other, one tank being filled while the other tank was being emptied. Corrections were made for variation of the temperature of the water. Corrections were also made for condenser leak-

age which amounted to only about 0.05 per cent. This leakage was determined by titration. The rotating standard watthour meters which were used for measuring the output were calibrated several times throughout the tests by the Electrical Testing Laboratories, New York City. It is believed that both the input and output were measured within an accuracy of 0.5 per cent. All

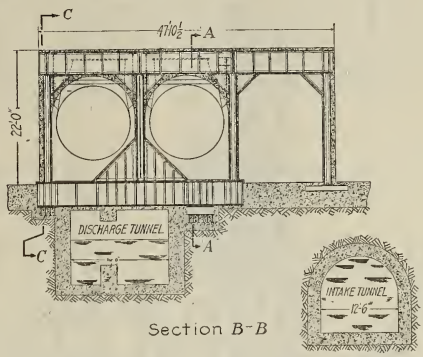

Fig. 8 Side Elevation of Foundation Steelwork

thermometers and gages were also calibrated in the usual manner. The specific gravity of the mercury used in the vacuum columns was determined and corrections made accordingly. The readings obtained from the mercury columns were further corrected for menis-

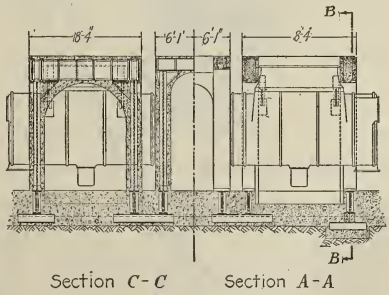

Fig. 9 End Elevation of Foundation Steelwork

cus, temperature and barometer reading. The barometer reading was obtained from the local U. S. Weather Bureau.

26 Most of the tests were of three hours duration. With the exception of a few special tests, the turbine was operated under normal conditions, in so far as type of load was concerned. The load was controlled from the switchboard through the remote

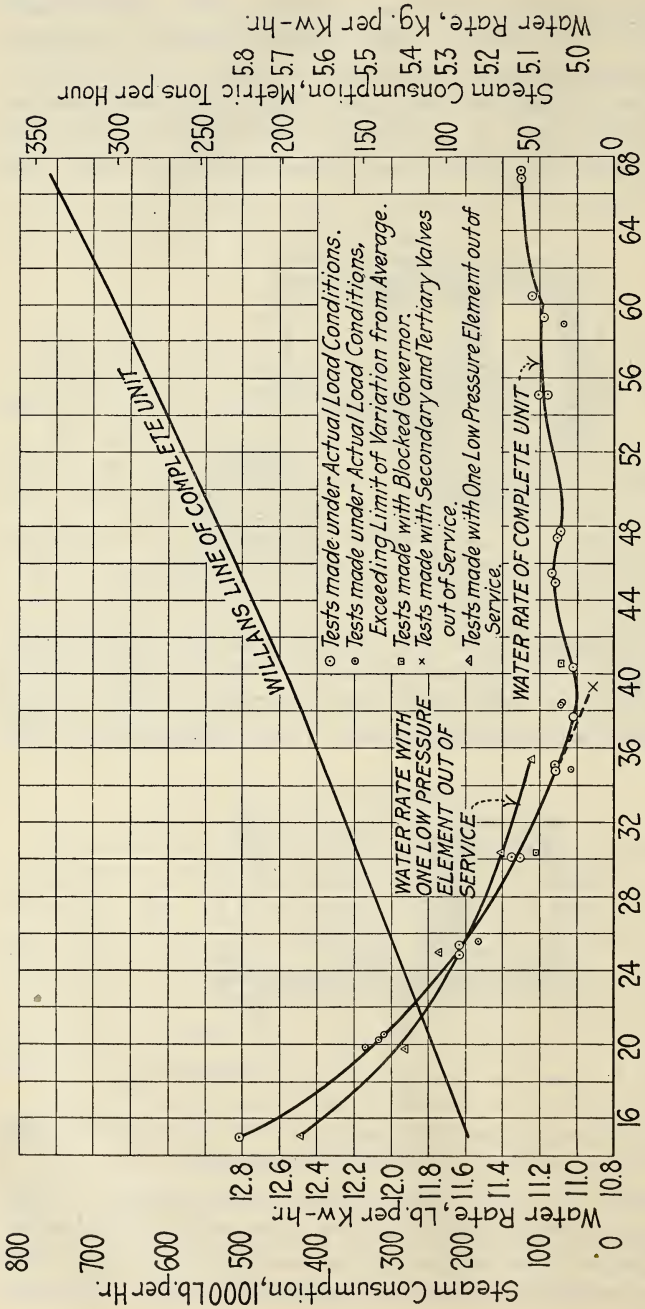

FIG. 11 WATER-RATE CURVES AND WILLANS LINE FOR 60,000-KW. TURBINE

Steam pressure, 220 lb. per sq. in. (15.47 kg. per cm.) abs.; superheat, 150 deg. fahr. (65.6 deg. cent.) vacuum, 29 in. Hg. (736.6 mm.) referred to a 30-in. (762 mm.) barometer; water rate factor 0.984. Curves based on tests Nos. 2 to 6 and 17 to 45, inclusive (excluding tests showing water rates differing from average curves by amounts exceeding 0.5 per cent, special tests with blocked governor, and test with secondary and tertiary valves out of service).

governor-control system provided for that purpose. This method of controlling the load subjected the turbine to the full swings of the railroad load.

27 Two series of tests were run, one with the complete unit in service and the other with the high-pressure and one low-pressure turbine in service.

28 Table 1 gives the numerical results of the turbine tests, while the performance is shown graphically in Figs. 11, 12, and 13. Fig. 11 gives the total steam consumption and water rate of the unit, while Fig. 12 gives the thermal and Rankine-cycle efficiencies.

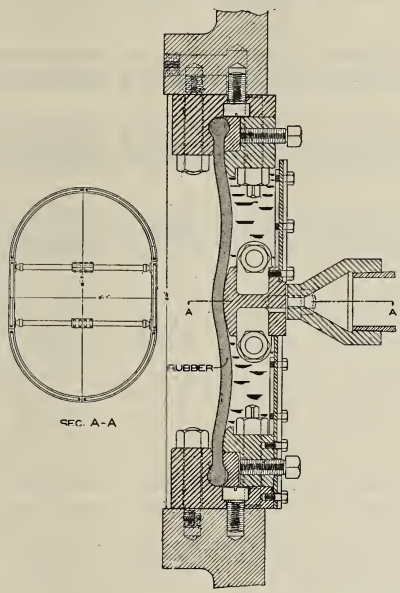

Fig. 10 Detail of Rubber Expansion Joint

From these curves it will be seen that the lowest water rate obtained while operating the complete unit under normal conditions was 11.00 lb. per kw-hr., while the highest Rankine and thermal efficiencies obtained were 76.0 per cent., and 25.1 per cent., respectively. With the high-pressure and one low-pressure turbine in service the lowest water rate was 11.25 lb. per kw-hr.

29 It will also be noticed in Fig. 11 that the water rate of the unit when the high-pressure and one low-pressure turbine is in service is better than that for the complete unit at loads below 25,000-kw.

TABLE 1 SUMMARY OF TURBINE TEST RESULTS

Test No.	Load Average, net kw.	Duration, hr.	Steam Pressure: Throttle, abs. lb. per sq. in.	Steam Pressure: Throttle, abs. kg. per sq. cm.	Steam Pressure: Exhaust vacuum[1] in. Hg.	Steam Pressure: Exhaust vacuum[1] mm. Hg.	Steam Temperatures at Throttle deg. fahr.	Steam Temperatures at Throttle deg. cent.	Superheat deg. fahr.	Superheat deg. cent.	Water: Actual total lb.	Water: Actual total kg.	Water: Corrected total[2] lb.	Water: Corrected total[2] kg.
8	15,038	3	230.0	16.17	29.11	739.4	539.0	281.7	145.2	80.7	569,704	258,412	578,085	262,214
36	20,231	3	219.5	15.43	28.61	726.7	522.2	272.3	132.5	73.6	769,254	348,926	731,925	331,994
32	20,534	3	221.4	15.57	28.56	725.4	519.1	270.6	128.6	71.4	784,970	356,054	741,357	336,272
43	24,928	3	220.5	15.50	28.47	723.1	520.4	271.3	130.3	72.4	924,420	419,308	869,796	394,531
34	25,274	3	219.0	15.40	28.45	722.6	516.6	269.2	127.1	70.6	940,025	426,386	881,487	399,834
26	30,102	3	221.7	15.59	28.44	722.4	531.9	277.7	141.3	78.5	1,075,971	488,045	1,023,660	464,322
30	30,146	3	222.2	15.62	28.40	721.4	522.2	272.3	131.4	73.0	1,087,750	493,393	1,023,465	464,233
40	34,648	3	223.2	15.69	28.28	718.3	523.5	273.0	132.3	73.5	1,233,880	559,676	1,155,531	524,137
28	34,798	3	221.0	15.54	28.38	720.8	524.3	273.5	134.0	74.4	1,229,740	557,798	1,161,006	526,620
45	37,605	3	221.6	15.58	28.15	715.0	522.0	272.2	131.5	73.1	1,337,720	606,776	1,243,581	564,076
17	40,332	3	219.4	15.42	28.27	718.1	528.2	275.6	138.5	76.9	1,414,619	641,657	1,333,596	604,906
37	44,910	3	222.0	15.61	28.08	713.2	528.0	275.5	137.3	76.3	1,603,180	727,186	1,496,472	678,784
31	45,436	3	218.9	15.39	28.03	712.0	524.7	273.7	135.2	75.1	1,631,650	740,100	1,517,100	688,145
39	47,407	3	219.4	15.43	27.93	709.4	522.6	272.5	132.9	73.8	1,709,612	775,463	1,578,171	715,843
27	47,647	3	220.4	15.49	28.03	712.0	523.9	273.3	133.8	74.3	1,703,985	772,911	1,584,600	718,759
41	55,059	3	218.9	15.39	27.76	705.1	545.2	285.1	155.8	86.6	1,967,280	892,338	1,844,070	836,452
35	55,063	3	220.0	15.47	28.07	713.0	539.8	282.1	149.9	83.3	1,949,370	884,214	1,849,497	839,273
44	59,283	3	219.4	15.42	27.44	697.0	542.3	283.5	152.6	84.8	2,157,430	978,589	1,987,863	901,674
33	60,456	3	218.7	15.38	28.09	713.5	545.1	285.0	155.7	86.5	2,128,240	965,348	2,036,007	923,512
42[4]	66,860	2	217.0	15.26	27.00	685.8	552.2	289.0	163.4	90.8	1,646,720	746,936	1,510,474	685,136
19[4]	67,168	2	216.2	15.20	27.33	694.2	547.3	286.3	158.8	88.2	1,634,762	741,512	1,517,722	688,423
20[5]	25,550	3	223.8	15.73	28.39	721.1	525.0	273.9	133.6	74.2	939,640	426,211	882,666	400,368
25[5]	34,853	3	222.5	15.64	28.40	721.4	521.7	272.0	130.8	72.7	1,222,737	554,621	1,153,530	523,221
38[5]	38,375	3	220.5	15.50	28.29	718.6	520.6	271.4	130.5	72.5	1,360,705	617,202	1,275,867	578,727
29[5]	38,544	3	220.5	15.50	28.31	719.1	518.5	270.3	128.4	71.3	1,365,685	619,461	1,280,529	580,834
18[5]	58,859	3	215.3	15.14	27.47	697.7	542.4	283.5	154.3	85.7	2,118,851	961,090	1,954,272	886,438
10[6]	19,902	2	228.2	16.04	28.45	722.6	544.1	284.5	151.0	83.9	506,383	229,690	482,362	218,794
13[6]	30,307	2	229.2	16.11	28.36	720.3	554.7	290.4	161.2	89.6	707,655	320,985	680,174	308,520
14[6]	40,547	3	224.0	15.75	28.33	719.6	530.5	276.9	139.0	77.2	1,416,240	642,392	1,348,530	611,677
22[7]	39,316	3	221.5	15.57	28.34	719.8	519.9	271.0	129.4	71.9	1,366,777	619,954	1,287,672	584,075
7[8]	No load	1	227.9	16.02	28.34	719.8	550.0	287.8	155.0	86.1	32,428	14,709		
3[9]	15,038	2	224.7	15.80	28.51	724.2	526.0	274.4	134.0	74.4	399,500	181,208	375,612	170,374
5[9]	19,842	2	222.6	15.65	28.35	720.1	527.0	275.0	136.0	75.6	507,630	230,256	473,934	214,972
2[9]	24,934	2	222.0	15.61	28.31	719.1	531.0	277.2	140.0	77.8	622,980	282,577	585,672	265,655
4[9]	30,168	2	218.0	15.33	28.04	712.2	531.0	277.2	142.0	78.9	740,880	336,056	688,486	312,290
6[9]	35,391	2	214.9	15.11	27.37	695.2	530.0	276.7	142.0	78.9	886,050	401,903	795,368	360,771

TABLE 1 (*Continued*) SUMMARY OF TURBINE TEST RESULTS

Test No.	Load Average, net kw.	Output, gross total, kw-hr.	Excitation,[3] kw-hr.	Output, net total, kw-hr.	Load, Average, Net Kw.			Water Rate Correction [2]		Rankine-cycle efficiency,[2] Per cent	Thermal efficiency,[2] Per cent
					Low-pressure element, right, kw.	High-pressure element, kw.	Low-pressure element, left, kw.	lb. per kw-hr.	kg. per kw-hr.		
8	15,038	45,579	465	45,114	3,955	7,128	3,955	12.81	5.810	65.3	21.5
36	20,231	61,171	479	60,692	5,301	10,459	4,471	12.06	5.470	69.3	22.9
32	20,534	62,082	480	61,602	5,195	10,986	4,353	12.03	5.456	69.5	22.9
43	24,928	75,271	488	74,783	6,706	12,240	5,982	11.63	5.274	71.9	23.7
34	25,274	76,319	497	75,822	7,026	12,511	5,737	11.63	5.274	71.9	23.7
26	30,102	90,814	508	90,306	8,128	14,750	7,224	11.34	5.144	73.6	24.3
30	30,146	90,941	503	90,438	7,989	14,741	7,416	11.32	5.132	73.9	24.4
40	34,648	104,466	523	103,943	9,805	16,250	8,593	11.12	5.044	75.2	24.8
28	34,798	104,916	521	104,395	9,500	16,390	8,908	11.12	5.044	75.2	24.8
45	37,605	113,354	538	112,816	10,567	17,449	9,589	11.02	4.998	75.9	25.0
17	40,332	121,532	535	120,997	10,567	18,149	11,616	11.02	4.998	75.9	25.0
37	44,910	135,297	568	134,729	13,473	18,683	12,754	11.11	5.039	75.3	24.8
31	45,436	136,874	566	136,308	13,267	19,129	13,040	11.13	5.048	75.1	24.8
39	47,407	142,796	576	142,220	14,222	19,769	13,416	11.10	5.035	75.3	24.8
27	47,647	143,514	574	142,940	14,533	19,535	13,579	11.09	5.030	75.4	24.9
41	55,059	165,777	600	165,177	17,564	20,867	16,628	11.16	5.062	74.9	24.7
35	55,063	165,792	603	165,189	17,675	20,703	16,685	11.20	5.080	74.6	24.6
44	59,283	178,474	625	177,849	18,971	21,876	18,436	11.18	5.071	74.8	24.7
33	60,456	181,996	628	181,368	19,708	22,006	18,742	11.23	5.094	74.4	24.6
42 [4]	66,860	134,159	440	133,719	22,866	22,265	21,729	11.30	5.125	74.0	24.4
19 [4]	67,168	134,785	449	134,336	22,971	21,763	22,434	11.30	5.125	74.0	24.4
20 [5]	25,550	77,145	495	76,650	6,822	12,775	5,953	11.52	5.225	72.6	23.9
25 [5]	34,853	105,080	522	104,558	8,888	16,415	9,550	11.03	5.003	75.8	25.0
38 [5]	38,375	115,666	541	115,125	11,052	17,192	10,131	11.08	5.026	75.4	24.9
29 [5]	38,544	116,168	535	115,633	10,638	17,499	10,407	11.07	5.021	75.5	24.9
18 [5]	58,859	177,198	620	176,578	19,012	21,483	18,364	11.07	5.021	75.5	24.9
10 [6]	19,902	40,117	314	39,803	4,916	10,210	4,776	12.12	5.498	69.0	22.8
13 [6]	30,307	60,942	329	60,613	7,940	14,608	7,759	11.22	5.089	74.5	24.6
14 [6]	40,547	122,181	540	121,641	11,596	17,841	11,110	11.09	5.030	75.4	24.9
22	39,316	118,480	532	117,948	11,009	17,810	10,497	10.92	4.953	76.6	25.3
7	No load	.....	154	.....	.....	.....	.....	.....	.....	.....	.....
3 [9]	15,038	30,292	215	30,077	10,151	4,887	.....	12.49	5.665	66.9	22.1
5 [9]	19,842	39,907	222	39,685	13,433	6,409	.....	11.94	5.416	70.0	23.1
2 [9]	24,934	50,104	236	49,868	16,955	7,979	.....	11.74	5.324	71.2	23.5
4 [9]	30,168	60,579	242	60,337	20,424	9,744	.....	11.41	5.175	73.3	24.2
6 [9]	35,391	71,044	262	70,782	24,597	10,794	.....	11.24	5.098	74.4	24.5

NOTES

[1] Referred to 30 in. (762 mm.) barometer at 58.1 deg. fahr. (14.5 deg. cent.).

[2] As corrected to 220 lb. per sq. in. (15.47 kg. per sq. cm.) abs. pressure, 150 deg. fahr. (65.6 deg. cent.) superheat 29 in. Hg. (736.6 mm.) vacuum ref'd. to 30 in. (762 mm.) barometer at 58.1 deg. fahr. (14.5 deg. cent.).

[3] Including rheostat loss.

[4] Tests Nos. 19 and 42 made with hand-operated quaternary valve wide open.

[5] Tests Nos. 18, 20, 25, 29 and 38 exceeding limit of variation from average, discarded.

[6] Tests Nos. 10, 13 and 14 made with operating governor adjusted to limit maximum loads to values shown in second column.

[7] Test No. 22 made with secondary and tertiary valves out of service.

[8] Test No. 7 made without load on generators, with generator field currents adjusted to give normal voltage on open circuit.

[9] Tests Nos. 2, 3, 4, 5 and 6 made on high-pressure and right-hand low-pressure elements only, with left-hand low-pressure element out of service.

30 A few tests were run with the governor blocked so that the unit operated under a steady load. These tests are shown in Figs. 11 and 12 by the small squares. It will be seen that there is no improvement in the efficiency under the steady load conditions at the loads selected for these tests. The test shown by the cross in Figs. 11 and 12 was run with the secondary and tertiary valves out of commission, while the load was adjusted so that the primary

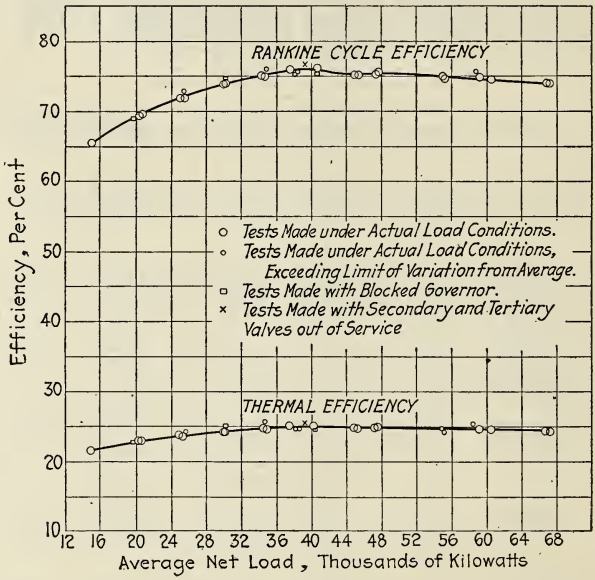

FIG. 12 RANKINE-CYCLE AND THERMAL-EFFICIENCY CURVES FOR 60,000-KW. TURBINE

Steam pressure, 220 lb. per sq. in. (15.47 kg. per sq. cm.) abs.; superheat, 150 deg. fahr. (65.6 deg. cent.); vacuum, 29 in. Hg. (736.6 mm.) referred to a 30-in. (762 mm.) barometer. Curves based on tests Nos. 17 to 45, inclusive (excluding tests showing water rates differing from average curves by amounts exceeding 0.5 per cent, special tests with blocked governor and test with secondary and tertiary valves out of service).

valve was wide open. As is to be expected, the lowest water rate was obtained while operating in this manner. Test No. 7 was run without any load on the generator, but with the field excited to give normal voltage on open circuit. The actual steam consumption per hour during this test amounted to 32,428 lb. No attempt was made to correct the steam consumption during this test, due to the load being outside the vacuum correction curves.

31 Fig. 13 shows the variation of steam pressure with the

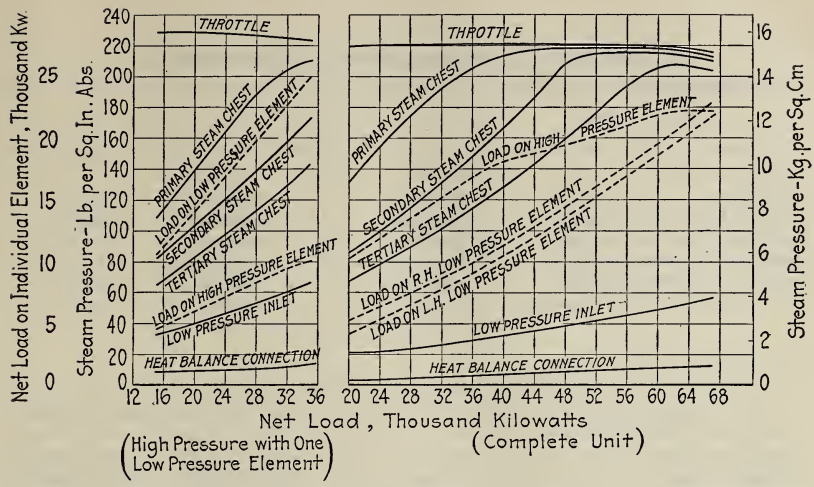

FIG. 13 VARIATION OF STEAM PRESSURE WITH THE LOAD AT VARIOUS POINTS THROUGHOUT THE TURBINE

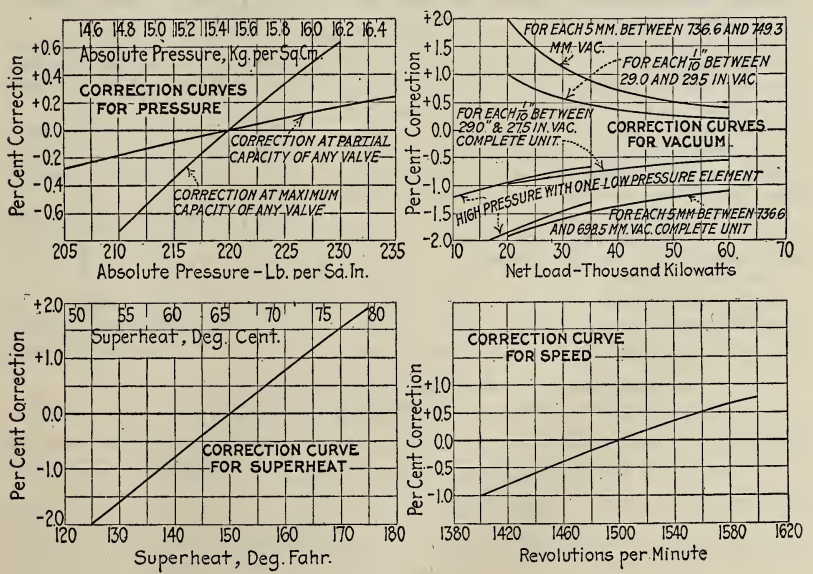

FIG. 14 CURVES FOR CORRECTING STEAM CONSUMPTION TO THE STANDARD CONDITIONS

TABLE 2 SUMMARY OF CONDENSER TESTS

Number of Test		3		1	
Duration of test	Hours.	3		3	
Barometer [1]	In. Hg.	30.17		30.29	
Load on turbine, average	Gross kw.	20,200		40,100	
Superheat at turbine throttle	Deg. fahr.	108		118	
Steam pres., turbine throttle, abs.	Lb. per sq. in.	232		228	
Temp. of circulating water in	Deg. fahr.	64.4		65.0	
Air leakage per min. (both units)	Cu. ft. at 60 deg. fahr.	18.0		12.7	
Number of air pumps in service		2		2	
Condenser Unit		Left	Right	Left	Right
Average load, steam per hr.	Lb.	119,333	132,667	224,333	236,333
Vacuum, average [2]	In. Hg.	28.80	29.01	28.81	28.82
Temp. corresponding to vacuum	Deg. fahr.	84.8	78.8	84.5	84.2
Temp. primary heater discharge	Deg. fahr.	77.7	75.1	79.4	78.8
Temp. hotwell discharge	Deg. fahr.	71.0	69.6	75.9	75.5
Temp. rise in primary heater	Deg. fahr.	6.7	5.5	3.5	3.3
Temp. of circulating water out	Deg. fahr.	67.6	67.9	70.5	70.9
Temp. rise of circulating water	Deg. fahr.	3.2	3.5	5.5	5.9
Circulating water (two pumps running)	Gal. per min.	69,200	70,600	74,000	72,700
Heat transferred per hour:	1000 B.t.u.				
Primary heater section		799.53	729.67	785.17	779.90
Condensing section		110,800	123,530	203,380	214,400
Entire condenser surface		111,600	124,260	204,170	215,180
Heat transf'd. per hr. per unit surf.:	B.t.u. per sq. ft.				
Primary heater section		333	304	327	325
Condensing section		2,328	2,595	4,273	4,504
Entire condenser surface		2,232	2,485	4,083	4,304
Mean temp. difference (log.):	Deg. fahr.				
Steam to condensate		10.07	6.04	6.70	6.92
Steam to circulating water		18.73	12.58	16.59	16.06
Heat transf'd. per hr. per unit surface per deg. M. T. D.:	B.t.u. per sq. ft.				
Primary heater section		33	50	49	47
Condensing section		124	206	257	280

[1] Barometer, in Hg. (mm. Hg.) at 58.1 deg. fahr. (14.5 deg. cent.)

[2] Vacuum, in Hg. at 58.1 deg. fahr. (mm. Hg. at 14.5 deg. cent.) ref'd. to 30 in. (762 mm. barometer.

TABLE 2 (*Continued*) SUMMARY OF CONDENSER TESTS

2			3		1		2	
3		Hours.	3		3		3	
30.15		Mm. Hg.	766.3		769.4		765.8	
60,233		Gross Kw.	20,200		40,100		60,233	
121		Deg. cent.	60.0		65.5		67.2	
222		Kg. per sq. cm.	16.31		16.03		15.61	
63.9		Deg. cent.	18.0		18.3		17.7	
14.8		Cu. dm. at 15.5 deg. cent.	510		360		419	
2			2		2		2	
Left	Right		Left	Right	Left	Right	Left	Right
345,667	380,000	Kg.	54,128	60,176	101,755	107,198	156,791	172,364
28.55	28.57	Mm. Hg.	731.5	736.9	731.8	732.0	725.2	725.7
90.7	90.3	Deg. cent.....	29.3	26.0	29.2	29.0	32.6	32.4
83.7	84.0	Deg. cent.....	25.4	23.9	26.3	26.0	28.7	28.9
80.1	79.8	Deg. cent.....	21.7	20.9	24.4	24.2	26.7	26.6
3.6	4.2	Deg. cent.....	3.7	3.0	1.9	1.8	2.0	2.3
72.9	73.3	Deg. cent.....	19.8	19.9	21.4	21.6	22.7	22.9
9.0	9.4	Deg. cent.....	1.8	1.9	3.1	3.3	5.0	5.2
70,500	74,100	Lit. per min....	262,000	267,300	280,100	275,200	266,900	280,500
		1000 kg-cal.						
1,244.4	1,596.0	....	201.48	183.88	197.86	196.54	313.59	402.19
316,980	348,350	.....	27,922	31,130	51,252	54,029	79,879	87,784
318,220	349,940		28,123	31,314	51,451	54,225	80,191	88,185
		Kg-cal. per sq. m.						
519	665	.....	903	824	887	881	1,408	1,803
6,659	7,318	.....	6,314	7,038	11,588	12,224	18,059	19,846
6,634	6,999	.....	6,053	6,735	11,073	11,672	17,259	18,981
		Deg. cent.....						
8.67	8.22	.....	5.59	3.36	3.72	3.84	4.82	4.57
21.98	21.35	.....	10.41	6.99	9.22	8.92	12.21	11.86
		Kg-cal. per sq. m.						
60	81	.....	161	245	239	229	293	395
303	343	.....	606	1,007	1,256	1,370	1,480	1,675

load at various points throughout the turbine. The curves in Fig. 14 were used for correcting the steam consumption to the standard conditions of 220 lb. absolute steam pressure, 150 deg. fahr. of superheat, 29 in. Hg. of vacuum and 1,500 r.p.m. These curves were furnished by the manufacturer, however the vacuum curves were checked to some extent by actual tests.

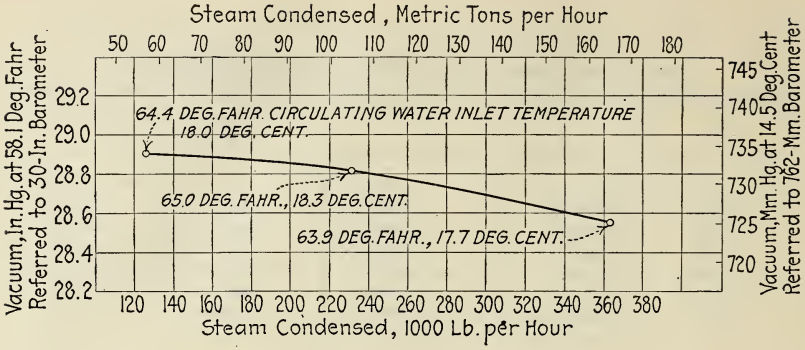

FIG. 15 CURVE SHOWING VACUUM OBTAINED DURING CONDENSER TEST

Based on tests Nos. 1, 2 and 3. Circulating-water inlet temperature varied from 63.9 deg. fahr. to 65.0 deg. fahr.

WATER-RATE FACTOR

32 In order to make an absolute and numerical comparison of the flatness of water-rate curves for different turbines, an expression has been used which is termed the "water-rate factor." Expressed mathematically,

$$F = \frac{MW}{C}$$

in which

F = water-rate factor
M = minimum water rate obtained
W = mean load between one-half rated load and maximum load, which is the usual operating range
C = average total steam consumption per hour covering the range between one-half rated load and maximum load.

Using the curves given in Fig. 11, substitutions may be made as follows:

$$F = \frac{11.00 \times 50{,}000}{558925.} = 0.984$$

CONDENSER TESTS

33 Separate tests were run in order to determine the performance of the condenser. The results of these tests are shown in Table 2, while the curve given in Fig. 15 shows the relation between load and vacuum for the entire unit.

No. 1853

STEEL FOR FORGE WELDING

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In this paper the principal factors — method of manufacture, chemical composition, fluxing quality, susceptibility to heat and welding temperature — affecting the welding quality of steel are discussed and the average results of 80 tests made on forge welds of hammer-welded pipe are compared with the original material. In addition it is stated that tests have demonstrated that both steel not over 0.15 per cent carbon and minimum tensile strength of 47,000 lb. per sq. in. and that not over 0.20 per cent carbon and minimum tensile strength of 52,000 lb. per sq. in., are satisfactory for forge welding of pipe lines, penstocks, tank-car work and similar construction but that the former is best adapted for welded parts of boilers and pressure vessels. In conclusion the writer believes that the most important consideration to produce uniformly good results in the forge welding of steel, is suitable material, well-trained operators and adequate facilities for the control of operations.

An appendix is devoted to a presentation of the Tentative Specifications for Steel Plates for Forge Welding of the American Society for Testing Materials as revised in 1921.

THE welding quality of steel, and the strength and reliability of such welds, depend on a number of factors, which include principally: method of manufacture, composition of the metal, susceptibility to heat, fluxing quality, the mechanical appliances for handling and controlling the work, and the skill of the operator. There are so many factors present affecting the results that it is often difficult to determine which of these predominates in any particular case. This paper discusses particularly the characteristics of steel for forge welding, with brief reference to other factors which enter the problem.

MATERIAL AND WORKMANSHIP

2 *Method of Manufacture.* Wrought iron is most easily welded, probably on account of the presence of about one and one-half per cent of easily fusible cinder, which enables the metal to be welded at a comparatively low temperature and protects it from injurious oxidation at high temperature. For this reason wrought

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iron can usually be welded without much difficulty, but on account of the presence of this cinder internal defects such as laminations and blisters are more likely to occur after the metal has been brought up to the welding heat. What we term " soft welding steel " may be made by the bessemer or open-hearth process and should be made especially for this purpose, i. e., it should have, as far as possible, sufficient of the characteristics of wrought iron to readily form a " welding scale " at the lowest possible temperature. Very highly refined open-hearth steels, " ingot iron " or electric steel, are, as a rule, lacking in this respect and so far have not shown as good welding quality as soft welding steel or wrought iron. Possibly these may be improved in this respect, but while these metals possess many advantages for other purposes, they do not at present appear to be so well adapted for forge welding.

3 *Composition.* It is well known that comparatively small quantities of nickel, chromium and silicon interfere seriously with welding. Each of these should be under 0.05 per cent. Carbon has a lesser effect and should preferably be low, certainly under 0.30 per cent for any kind of forge welding. The higher the carbon, the lower the melting and burning point of the steel. By the burning point we mean the temperature at which the grain growth has increased to such a degree as to cause actual disintegration and intergranular oxidation of the metal. Sulphur under 0.05 per cent is not harmful and under certain conditions more may be present without injurious results. Phosphorus up to bessemer limits is beneficial to welding.

4 *Self-Fluxing Quality.* On heating iron or steel above 1500 deg. fahr. an oxide scale is formed. The relation between the fusibility of the oxide scale to the temperature at which the metal " burns " is one of the most important factors determining suitability of the metal for welding. This scale consists usually of the magnetic oxide of iron (Fe_3O_4) with a certain percentage of " sonims "[1] from the iron (MnO, P_2O_5, SiO_2, etc.) which tend to make the scale more fusible. The method of manufacture and composition of the steel have much to do with the formation of a suitable welding scale. The range of temperature between the melting point of the scale and the burning point of the metal is about 100 deg. fahr. in good welding steel and distinguishes this

[1] Solid non-metallic impurities in steel, H. D. Hibbard, Trans. A. I. M. E., vol. xli, p. 803 (1910).

class of steel probably more than any other property. In fact, it is this self-fluxing quality which makes possible the commercial welding of iron and steel. Artificial fluxes, such as borax, may be used to lower the melting point of the scale in welding small parts but at present fluxes are not practicable to apply satisfactorily when working on large production. The fusion of the scale also affords the operator a definite indication of the welding heat, giving him close control over the operation.

5 *Susceptibility of Metal to Heat.* When normal wrought iron or steel is heated above the upper critical point (about 1750 deg. fahr. for soft steel) the grain grows at a rate depending on the temperature and time of heating. When a certain grain size is reached, a disintegration of the metal occurs with intergranular oxidation and the metal becomes " burnt ". When this occurs, the metal is both red-short and cold-short and useless for most purposes. The actual temperatures at which iron or steel is burned depends as much on the protective character and fusibility of the welding scale as anything else. High-carbon steels are more susceptible to damage of this kind in welding than the same class of steel of lower carbon but the carbon is not the only factor, otherwise we might expect highly refined open-hearth steel or "ingot iron" to weld as easily as charcoal iron.

6 The large granular structure caused by exposure of the metal to welding temperature may be reduced to a fine structure (unless the metal has been excessively overheated) by a certain amount of mechanical forging applied while the metal is cooling or by reheating the metal to about 30 deg. fahr. above the upper critical point, followed by cooling in the air, which with soft steel may be comparatively rapid.

7 *Welding Temperature.* To produce intercrystalline union of two pieces of iron it is necessary that the clean surfaces be brought into close contact with a certain pressure. This is possible even at normal temperature with application of sufficient pressure in the case of soft steel, or may easily be done at a temperature slightly above the fusing point of the scale with comparatively little pressure, or at a lower temperature if the fusion point of the scale is lowered by the use of artificial fluxes, such as borax. So that the most favorable temperature for welding depends on the material and mechanical facilities. The usual temperature at which soft steel is found to weld satisfactorily ranges from 2500 to 2600 deg. fahr.

8 The skill and experience of the operator is, of course, a considerable factor in all welding. However, this is offset in forge welding to some extent by the facilities given him for controlling the heat and the work.

RESULTS OF TESTS

9 A number of tests of forge welds (80 in all) made on two rings cut from the ends of hammer-welded pipe about ½ in. thick, compared with the original material taken from the same pipe, 90 deg. from the weld, gave results which are summarized as follows:

Material Away from Weld — Average transverse tensile test.	
Elastic limit, lb. per sq. in.	32150
Ultimate strength, lb. per sq. in.	52790
Elongation in 8 in., per cent	29.7
Reduction, per cent	58.6
Efficiency of Weld — Test pieces machined to uniform thickness.	
Average of all tests (80 tests), per cent	92.7
Average at extreme end (40 tests), per cent	90.3
Average 2 in. or more away from end (40 tests), per cent	95.0
Minimum at extreme end, per cent	69.0
Minimum 2 in. or more away from end, per cent	82.3

10 The above steel before welding ranged in tensile strength from about 47,000 to 62,000 lb. per sq. in.— most of it being under 57,000 lb. and under 0.16 per cent carbon.

SPECIFICATIONS

11 This brings us to the question of specifications for steel best suited for forge welding. While skillful operators can undoubtedly make a good job of most steels when the carbon does not exceed that of flange steel,[1] it seems desirable, everything considered, to limit the carbon to about 0.15 per cent for important parts where life and valuable property are at stake and a high efficiency of strength of weld is desired.

12 The present A.S.T.M. specification (A78–21–T) for forge-welding steel (see Appendix) calls for steel of not over 0.18 per cent carbon having a minimum tensile strength of 50,000 lb. per sq. in. A.S.T.M. Sub-Committee II of Committee A-1 now have under

[1] The term "flange steel" is used in the usual sense, referring to steel of flanging quality ranging between 0.15 and 0.25 carbon, although this quality is not confined to steel of these limits.

consideration substituting for this two grades of steel having the following chemical and physical properties:

	Grade A	Grade B
Chemical Composition		
Carbon, per cent[1]	not over 0.15	not over 0.20
Manganese, per cent	0.35 to 0.60	0.35 to 0.60
Phosphorus, per cent	0.04	0.04
Sulphur, per cent	0.05	0.05
Physical Tests		
Tensile strength, lb. per sq. in.	not under 47000	not under 52000
Yield point, lb. per sq. in.	not under 25000	0.5 tensile strength
Elongation in 8 in., per cent	not under 26	not under 24

[1] For plates over ¾ in. thick, 0.02 additional carbon is permissible.

13 Steel of both grades has been forge-welded and used in large quantities with an assumed weld efficiency of 90 per cent. The tests we have made indicate that this figure is warranted for pipe lines, penstocks, tank-car work and similar construction for which "Grade B" may be used. A somewhat lower efficiency or higher factor of safety should, of course, be used for boilers and Class A unfired pressure vessels for which "Grade A" is best adapted.

BOILER-CODE REQUIREMENTS

14 With respect to steel for forge welding, Part I, Section I, Par. 186 of the Boiler Code requires that:

The ultimate strength of a joint which has been properly welded by the forging process, shall be taken as 28,500 lb. per sq. in., with steel plates having a range in tensile strength of 47,000 to 55,000 lb. per sq. in. Autogenous welding may be used in boilers in cases where the strain is carried by other construction which conforms to the requirements of the Code and where the safety of the structure is not dependent upon the strength of the weld.

Section III, paragraph L-29 reads:

The ultimate strength of a joint which has been properly welded by the forging process, shall be taken as 28,500 lb. per sq. in., with steel plates having a range in tensile strength of 45,000 to 55,000 lb. per sq. in. Autogenous welding may be used in boilers in cases where the strain is carried by other construction which conforms to the requirements of the Code and where the safety of the structure is not dependent upon the strength of the weld.

The proposed section on unfired pressure vessels with reference to forge weldings, Pars. 5 and 8, reads:

The ultimate strength of a joint which has been properly welded by the forge process shall be taken as 65 per cent of the tensile strength of the plate.

This weld efficiency seems rather low for Class A vessels and should be still higher for Class B vessels.

15 In Pars. 2 and 3 of Sections I and III, firebox and flange steel are specified *for all parts* of the boiler. There seems to be a conflict in these specifications between the requirements for steel which may be forge-welded, although apparently the intention is to use a steel of lower carbon for this purpose. This would seem to be in line with the best experience, but inasmuch as flange steel has apparently been successfully used for some time in forge-welded boiler construction where part of the stress is carried by riveted straps, there would seem to be no reason for not continuing this practice when the weld is so reinforced.

FINISHING

16 After the weld is made, internal strains remain in the metal which should be released by annealing. This may be done by heating the piece uniformly to a red heat (about 1500 deg. fahr.) and allowing to cool in the air. Any objectionable amount of distortion which has occurred in the welding operation should be removed, preferably while the piece is at an annealing heat; otherwise it should be reformed and then annealed. Nothing has been said as to the method of welding, scarfing, preparation of the plate, or fuel to be used, as these vary considerably and good results have been obtained with widely different methods of working. Some operators prefer one form of scarfing, others none at all. Some use roller welding machines, but the majority use power hammers.[1] Good welding has been done with coke fire, producer gas, natural gas and water gas, the last being best adapted for forge-welding on a large scale.

17 To produce uniformly a high weld efficiency, the most important considerations are: suitable material, well-trained operators, and adequate facilities for control of the work.

[1] Details of mechanical appliances for hammer welding in modern American plants will be found in articles by E. F. Thum in *Chemical & Metallurgical Engineering,* September 21, 1921, October 19, 1921, and November 16, 1921.

APPENDIX

TENTATIVE SPECIFICATIONS FOR STEEL PLATES FOR FORGE WELDING[1]

Serial Designation: A 78–21 T. American Society For Testing Materials.

18 These specifications apply to steel plates for forge welding for car tanks and for similar construction.

I MANUFACTURE

19 The steel shall be made by the open-hearth process.

II CHEMICAL PROPERTIES AND TESTS

20 (*a*) The steel shall conform to the following requiremens as to chemical composition:

Carbon	for plates ¾ in. or under in thickness, per cent	not over 0.18
	for plates over ¾ in. in thickness, per cent	not over 0.20
Manganese, per cent		0.40 — 0.60
Phosphorus, per cent		not over 0.04
Sulphur, per cent		not over 0.05

(*b*) The composition of steel for forge-welding plates should preferably be free from silicon, nickel or chromium. Where these elements are present the maximum quantity of any one shall not exceed 0.05 per cent.

21 An analysis of each melt of steel shall be made by the manufacturer to determine the percentages of carbon, manganese, phosphorus and sulphur. This analysis shall be made from a test ingot taken during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative, and shall conform to the requirements specified in section 20.

22 An analysis may be made by the purchaser from a broken tension test specimen representing each melt. The chemical composition thus determined shall conform to the requirements specified in section 20.

III PHYSICAL PROPERTIES AND TESTS

23 (*a*) The material shall conform to the following minimum requirements as to tensile properties:

Tensile strength, lb. per sq. in.	50 000
Yield point, lb. per sq. in.	0.5 tens. str.
Elongation in 8 in., per cent	1 500 000 / Tens. str.

[1] Issued, 1919; Revised, 1920, 1921.

(*b*) The yield point shall be determined by the drop of the beam of the testing machine.

24 (*a*) For material over $\frac{3}{4}$ in. in thickness, a deduction from the percentage of elongation specified in section 23 (*a*) of 0.25 per cent shall be made for each increase of $\frac{1}{32}$ in. of the specified thickness above $\frac{3}{4}$ in., to a minimum of 20 per cent.

(*b*) For material under $\frac{5}{16}$ in. in thickness, a deduction from the percentage of elongation in 8 in. specified in section 23 (*a*) of 1.25 per cent shall be made for each decrease of $\frac{1}{32}$ in. of the specified thickness below $\frac{5}{16}$ in.

25 The test specimen shall bend cold through 180 deg. flat on itself without cracking on the outside of the bent portion.

26 (*a*) Test specimens shall be prepared for testing from the material in its rolled condition.

(*b*) Test specimens shall be taken longitudinally and, except as specified in paragraph (*c*) shall be of the full thickness of material as rolled. They may be machined to the form and dimension illustrated in complete specification bulletin, or with both edges parallel.

(*c*) Test specimens for plates over $1\frac{1}{2}$ in. in thickness may be machined to a thickness or diameter of at least $\frac{3}{4}$ in. for a length of at least 9 in.

(*d*) The machined size for rectangular bend-test specimens may have the corners rounded to a radius not over $\frac{1}{16}$ in.

27 (*a*) One tension and one bend test shall be made from each melt; except that if material from one melt differs $\frac{3}{8}$ in. or more in thickness, one tension and one bend test shall be made from both the thickest and the thinnest material rolled.

(*b*) If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

(*c*) If the percentage of elongation of any tension-test specimen is less than that specified in section 23 (*a*) and any part of the fracture is outside the middle third of the gage length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV FINISH

28 The finished material shall be free from injurious defects and shall have a workmanlike finish.

V MARKING

29 The name or brand of the manufacturer and the melt number shall be rolled or stamped on all finished material. The melt number shall be legibly marked, by stamping if practicable, on each test specimen.

VI INSPECTION AND REJECTION

30 The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost,

all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

31 (*a*) Unless otherwise specified, any rejection based on tests made in accordance with section 22 shall be reported within five working days from the receipt of samples.

(*b*) Material which shows injurious defects subsequent to its acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

32 Samples tested in accordance with section 22, which represent rejected material, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

DISCUSSION

Stanley G. Child. The writer has written to a large number of welded-tank manufacturers with the view of determining the present actual practice in this industry.

The York Manufacturing Company states: "We have had no difficulty in welding standard steel tank plate, excepting with that which contains segregations or impurities."

The Frick Company states: "We do not favor a specified limitation of carbon content, but would suggest that the standard specification for structural and boiler steel as adopted by the Association of American Steel Manufacturers be made the standard."

The Struthers-Wells Company states: "We have been doing electric welding for seven years and gas welding for thirteen years and have never specified or used any other grades of steel plate than the standard grades of tank and flanged steel as made by the Association of Steel Manufacturers. As a concern doing as much or more welding than any in the United States we see no object in attempting to use a softer grade of steel."

The Baldwin Locomotive Works has welded pressure tanks for the past ten years, using the regular grade of boiler steel, without a single record of failure in service.

The William B. Pollock Company of Youngstown, Ohio, states that it is their practice to weld tank material to manufacturers' specification of steel 55,000 to 65,000 lb. tensile strength and carbon content of 0.15 to 0.25, with very satisfactory results.

The Coatesville Boiler Works state: "We have been welding steel plates of tank, flange and firebox grades for a number of years. The steel is all furnished in accordance with the specifications of the Association of Steel Manufacturers."

Henry Vogt Machine Company, Louisville, Ky., states: "We used flange steel of 55,000 to 65,000 lb. tensile, which has a carbon content of 0.15 to 0.25 and is very satisfactory. We have had no trouble with it whatsoever."

John Wood Mfg. Company of Conshohocken, Pa., states: "We use steel of maximum 0.15 carbon with no required tensile strength."

From the replies received it was evident that a large part of the pressure-tank practice of the present time was based on the use of ordinary grades of boiler and tank steel having a tensile strength of 55,000 to 65,000 lb. and governed by the specifications of the American Society for Testing Materials, the Association of American Steel Manufacturers, or the American Railway Association, which were practically identical and which called for rigid chemical and physical tests.

Some manufacturers were using a softer grade of steel for welded tanks with satisfactory results. This grade was equivalent to the A.S.T.M. specification A–78–21–T.

The writer believes that both of these grades of steel should be permitted for welding practice. While a number of experimental tests have been made to show the superiority of soft steels over the standard grades for welding, the cases are more or less special and are not representative of general tank practice. It is probable, for example, that welds made by the electric resistance autogenous process, without adding additional metal, should be restricted to a soft grade of steel as has been found desirable in the forge welding of steel pipe, which is lap-welded and has no metal added.

The Author. The references quoted by Mr. Child are not specific enough in that they do not distinguish between the requirements for fusion welding and forge welding or the use to which the welded vessel is to be put, all of which would seem to be important factors. It is quite likely that a different grade of steel than that used for forge welding will be found best adapted for fusion welding although the carbon limits may be the same.

In reply to a question on the hammer test it may be said that this was developed some eighteen years ago by the National Tube Company in connection with the manufacture of lap-welded steel

tubes and has been found to be of considerable service. The actual force of impact is not of so great importance although the force should be somewhat in proportion to the size of the piece tested. In the case of boiler tubes a two-pound hand hammer was used; later on pneumatic and mechanically operated hammers were developed on account of the danger to the operator using the hand hammer. In the case of the large forge-welded pipe no standard has been arrived at. The proposed code for unfired pressure vessels recommends a hammer weighing from 4 to 10 pounds. Any hammer which will send vibrations through a cylinder without denting the metal is sufficient. Such a blow on a vessel under static pressure will often cause fracture although the vessel shows no signs of failure under static pressure, indicating that the vibratory stress is better than a much higher static pressure. In regard to the tensile strength specifications of the present code, these are incomplete and more than the minimum tensile strength should be stated. The 47,000-pound minimum tensile mentioned in one part of the code should be adopted for "Class A" vessels which are welded but other parts of the code should be modified so as to be consistent with this specification.

In regard to the use of flange and firebox steel, it is unfortunate that such grades of steel as usually specified for boiler work happen to be just a little too high in carbon to be best suited for welding. It is, of course, possible to make good and sound welds of such steel with care but the Boiler Code Committee is in search of steels with which there is the least difficulty in welding and it was thought that even though another grade of steel was necessary this was better in order to get the best material for forge welding. There would seem to be sufficient tonnage required to warrant a special grade of weld steel. The minimum limit of 50,000 lb. tensile strength in A.S.T.M. specification A–78–21–T is a compromise between those who want higher tensile strength and those wanting good welding qualities.

No. 1854

NATIONAL LEADERSHIP

By Dexter S. Kimball, Ithaca, N. Y.
President of the Society

THE ultimate criterion by which any civilization will be judged will not be its armies or navies, its great edifices nor its wealth, but the degree of well-being — physical, mental and spiritual — that it bestows upon its people. This problem of universal well-being is most ancient and comes naturally from man's desire to enjoy the good things of life before he goes hence. The attainment of even an approximate realization of this desire was denied to all the old handicraft civilizations that preceded the present era. It does not appear to be possible to create by handicraft methods sufficient worldly comforts to support a great nation without having the larger part of the population work as menials with little or no mental development. The national ideals of all handicraft nations are, therefore, those of necessity and hold little hope of physical comfort this side of the grave.

THE PROBLEM OF UNIVERSAL WELL-BEING

But there has been born to us in these latter days a hope that this old desire may yet be realized, and this hope comes to us through the use of modern methods of production. Our civilization differs from those that have gone before, and from some that exist even today, only in one important particular. Our philosophy and our religions are built up for the most part of beliefs inherited from our forefathers, but our power to produce the necessities of life, to feed, clothe, and house the multitude, stands out as a thing apart and unlike anything that has as yet appeared on this earth so far as we have record. This power has come to us through the use of what we are pleased to call the "scientific

Presidential Address at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

method," by which we aim finally to replace the words "I think" with the words "I know" in all of our mundane activities. Our success in the use of this method in conquering unfriendly nature and turning her resources to our use has been most remarkable, and as it has become increasingly clear that physical, mental and spiritual well-being depend primarily upon the industrial background, the insistence that the benefit of modern industrial methods be more widely distributed has increased in like proportion. If I read the ideals of American democracy aright, we are committed in this country to an endeavor to secure universal well-being.

The benefits that have accrued to all classes of people in modern civilization are beyond question and are more apparent when comparison is made either with older forms of civilization, or even with existing handicraft nations. The general level of physical comfort and education as seen in modern nations is unquestionably higher than has ever before been attained. Nevertheless, no single class of people is satisfied with its position in the nation, and there is a deep feeling of unrest and discontent among those workers who actually produce the comforts of life. This is probably not new and undoubtedly has had its parallel in all older civilizations wherever large bodies of people dwelt closely together and where division of labor was at all a feature of industrial life. A multitude of reasons are advanced for this dissatisfaction and a multitude of remedies are offered for its mitigation, many of them having little or no bearing upon the real roots of the problem. It is obvious, however, from these suggested remedies that much if not most of these troubles have their roots in the industrial system under which we live — as might be expected, for industry is the very backbone of all existence. An examination of industrial laws preceding the present era and going back into remote antiquity indicates that modern industry has introduced few, if any, new problems. It has, however, accented and made more acute industrial problems that always arise where congregated industry and division of labor are factors in production.

CHANGES WROUGHT BY MODERN METHODS UPON INDUSTRIAL LIFE

Briefly, and without detail, the most important changes wrought by modern methods upon industrial life are as follows: The separation of agriculture and mechanic arts that began hundreds of years ago has been completed and widened. The agricul-

tural worker now depends upon others for his tools of production. Within the mechanic-arts group the actual worker has been separated almost entirely from the ownership of the tools of industry. A complex system of transportation independent of either of these older fields of occupation has been developed. And, lastly, both in transportation and the mechanic arts, division of labor has been extended to a degree unheard of by our ancestors and made possible only by the use of modern tools of industry. The resulting complexity of human relations coupled with the vast increase in population has accented the time-old problem of "what is mine and what is thine?" to a point where many people despair of a a solution and openly advocate a return to simpler methods and consequent simpler relations. The cry for justice, whatever that may be, is still abroad in the land, and we do not appear to have made much progress in attaining justice since the day when Plato described it as the essence of all good things.

QUALIFICATIONS OF THE ENGINEER FOR ATTACKING THE PROBLEM OF EFFECTIVE WEALTH DISTRIBUTION

This charge that modern civilization is a failure or at least no improvement upon former civilization, is, or should be, of peculiar interest to the engineer, using this term in its widest sense to include all industrial workers who use the scientific method. For modern civilization is largely what he has made it, and the civilization of the future will be largely what he wishes it to be. It is too much of course to claim that the engineer, unaided, can solve these difficult problems, but it is undoubtedly true that if he will direct his energies to the problem of the distribution of wealth as earnestly as he has devoted them to its production he can make a contribution to industrial economics that will be exceedingly helpful. The engineer, and he alone, has a direct and personal knowledge of the great industrial machine that he has created. Until recently he has entrusted the operation of this machine to others who often knew little about its refined mechanism. It is high time that he took charge and operated this complicated mechanism himself.

There would seem to be little doubt but that modern productive methods on the farm and in the factory will eventually produce sufficient wealth to make universal well-being possible if our methods of distributing this wealth can be made properly effective

to this end. Economists tell us that, as yet, we have not reached this point of production. It would seem, however, that even now if our energies were all bent more upon the production of necessities and less upon luxuries and pleasure-giving products, a great rise in the general level of well-being would occur. A superficial examination of every-day surroundings will convince any one of the tremendous amount of productive effort now wasted upon the things that profit us but little physically, mentally or spiritually. To say the least, the prospect of a higher level of well-being viewed from the standpoint of necessary production is very hopeful.

Furthermore, it should be noted that the efforts of the engineer are no longer confined to the design and actual production of commodities. The principles of mass production that have so greatly reduced the cost of commodities he has now quite fully developed. The wide application of the principles of mass production has been made possible by mass financing, the work of the financier. The engineer, in turn, is now developing the principles of mass management, and his contributions to the philosophy of management are already noteworthy. I believe most business men would be surprised to know to what a large extent the methods of the engineer have invaded their chosen field, and there can be little doubt but that the near future will see the engineering type of manager a predominant figure in all industrial administration.

This adventure into a broader field has brought the engineer for the first time face to face with the greater problem of human relations in industry and the distribution of wealth. In times past he has given little or no thought to these problems, but has assumed that they lay outside of his field. This assumption is no longer true, and whether he wishes to consider these problems or not, they confront him in this new field of industrial management. He cannot avoid the issue.

It will be freely conceded that the preparation of the engineer for these new duties is far from adequate, and two of his shortcomings stand out conspicuously. The first is his lack of knowledge of the economic principles of industry and commerce. It is true, of course, that the field of abstract economics is still far from being an exact science and many of its theories are controversial. But that is no excuse for a lack of knowledge, on the part of the industrial manager, of this important and growing subject which aims to replace ignorance and empiricism with accurately reasoned results. Furthermore, every engineer is by nature an economist.

The location of an industrial plant, the design of a great power house, the location of a railway are all problems in economics, and the tangible things with which economics is concerned are familiar objects to every engineer. This need in the equipment of the engineer is now fully recognized, and in a fair way to be remedied. All first-class engineering colleges now prescribe more or less economic study, and students of engineering recognize this study as a necessary part of their mental equipment.

Another important weakness in the mental processes of the engineer is his lack of knowledge of human nature and, worse still, his lack of sympathy with human problems. He is by nature a constructive individualist and usually impatient to obtain results; and, as a consequence, impatient of any obstruction, human or physical, that stands in the way of these results. And he is by nature a cave dweller given to introspection, unskilled in expressing his views verbally, and lacking usually in the companionable qualities that take strong hold upon subordinates. Of course there are many exceptions to these general criticisms, but in the main I believe they are true. This weakness, no doubt, is responsible for the failure of engineers as a class to take an interest in all civic matters and the reason why they have been, until very lately, a negligible factor in national affairs; though some of the present-day national problems are of especial interest to all men of this calling. But it would seem that this difficulty is, also, in a fair way to be overcome if one may judge by the great activity manifested lately by local and national engineering organizations in civic and national problems.

Over against these weaknesses of the engineer must be placed certain inherent qualities that strengthen his claim to industrial leadership. He is as a class essentially honest. His entire training and professional experience are such as to demand honesty of thought and purpose in all of his transactions. It is true, of course, that here also there are exceptions, but they are comparatively rare ones. Engineers as a class are honest, and if anything is needed today in management more than anything else it is this ancient virtue.

The engineer also often occupies a strategic position in industry which is of great importance in controversial matters. As the designer and planner of industrial enterprises he stands between capital and labor, and he is, not infrequently, better informed of the real difficulties at issue than are either sides of the controversy.

Furthermore, human progress must rest to a large extent upon a knowledge of the experiences of the race. This is true in all lines of human activity, and particularly true of a civilization that rests upon scientific achievement. The hope of the future lies largely in making accessible to all the great mass of accumulated experience that has come down to us. The engineer, of all men, has stood out preëminently in this respect as one who knows and uses classified knowledge to the fullest extent of his ability. The great leaders of the past had, for the most part, no conception of the importance of recorded experience. With few exceptions they took their knowledge and experience with them. The modern engineer has been the first to endeavor to reduce to written form the basic principles of industrial leadership, and his work in this field has already commanded worldwide attention.

And last, but most important, the engineer has command of the scientific method of attacking problems. With this he has practically solved the problem of production and is now rapidly rebuilding the field of administration. Is it unreasonable to suppose that he cannot make a great contribution to the problem of the distribution of wealth if he attacks this problem in the same whole-hearted manner that he has applied himself to these other fields of activity? There is indeed very good reason to believe that if he makes proper preparation he will bulk large in the industrial leadership of the future.

A NEW FORM OF INDUSTRIAL LEADERSHIP NEEDED

All thinking men who know anything about industry agree that we are sadly in need of a new form of industrial leadership, but the source of this new leadership does not seem to be clear. And it should be noted that future industrial leadership will be closely associated with political leadership, so closely are the problems of industry linked with political administration. History gives us an idea of what has occurred in the past and gives us some basis from which we can draw conclusions as to the leadership of the future. The first form of national leadership was the military type, the rule of the strong man. This extreme form of leadership still prevails in some countries, though it is hoped that it has seen its day and will soon disappear from the earth. It was natural that the military form of government should be adopted in the first factories and large industrial enterprises. And this principle

has certain inherent advantages that will undoubtedly prolong its use, though the modern tendency is to modify its more extreme features. The complexities of industry and the dependence that must now be placed upon expert advisers render any form of autocratic industrial government almost impossible.

In modern democracies military rule has been succeeded by what may be termed legal government. Today our social, political, and industrial activities are governed by a very comprehensive code of legal regulations that rest upon precedent and upon the opinions of eminent jurists sometimes affirmed and reaffirmed many times. These legal restrictions are a valuable guide since they embody the experience of the race with conditions that have occurred The lawyer has always rendered and is still rendering a great service to humanity in collecting public opinion and codifyng it for our guidance; and for this reason the lawyer has always been prominent in public affairs as an interpreter of justice as prescribed by precedent and usage. The legal method and the legal mind, however, are for the most part backward-looking in their processes. They aim to correct abuses as they arise in the body politic or in industrial life, and while the legal mind is of immense value to society, it cannot be said to have developed any philosophy of industrial life that holds out hope of a solution of these problems. An excellent illustration of our modified point of view as affecting the legal government of industry is to be found in our modern compensation laws. For many years all courts have held that a workman could secure compensation for injuries only if he could prove that he had not been guilty of negligence and that his fellow-workers had not been guilty of contributory negligence. Or in other words, the burden of proof was upon him to show that the blame lay upon the employer. This legal ruling had been confirmed repeatedly and it should be noted that, in general, the worker had to have recourse to law to secure compensation. Suddenly, almost, these old laws were swept from the statute books and the modern compensation acts passed which provide that workmen and their families shall not be made to suffer because of the dangers of industry. They practically affirm that accident compensation is a proper charge upon production and should be borne by the community interested and not by the unfortunate workman and his family who, usually, are not in a financial position to assume this load. This is clearly a step toward universal well-being and indicative, I believe, of other similar changes. There is, therefore, good

reason to believe that the legal type of mind is not likely to produce the much needed new industrial leadership. In fact, it is more than likely that the legal regulations of the future will not rest so much as heretofore upon ancient usage, but will be greatly modified in their viewpoint by the new industrial conditions.

There remain for consideration two other types of mind from which a new industrial leadership may come, namely, the business or financial type, and the scientific type. The business type of mind with the aid and counsel of the legal and engineering fraternities has built up modern industry as we now see it. There can be little doubt as to the ability of this type of mind to plan and direct the larger aspects of industry along existing lines. One weakness of this type of mind lies in its apparent inability to appreciate or rather to acknowledge that a new industrial day has dawned in which industry is being viewed more and more as the support of human life and not as a means of producing private, corporate, or state profits. It is rather reluctant to admit the philosophy of universal well-being as defined in the foregoing. Whether it can be made to appreciate this view is an open question.

Another weakness of this type of mind is its lack of technical knowledge of modern industry. As industry becomes more and more complex the business man is compelled to depend more and more upon the engineer for advice and knowledge. It is this very phase of modern industry that is rapidly forcing the role of industrial manager upon the engineer. The truth of these statements is shown in considering complex industrial problems like the twelve-hour shift, which is partly at least a technical problem. The best report upon this subject that has as yet appeared is the report of the committee appointed by the Federated American Engineering Societies. It is the best because it is based upon accurate technical knowledge of the industry and not upon speculative hearsay as most of these industrial reports are. The futility of any other kind of report is well illustrated in the Muscle Shoals controversy, which could be settled quickly and accurately by a properly chosen engineering commission. If the new industrial leadership comes, therefore, from the business world, the business type of mind, I believe, will have to be greatly modified.

Dean Inge, the noted London divine, reasoning along lines similar to the foregoing, comes to the conclusion that the only hope for new industrial leadership lies with the business type of mind on the ground that the scientific type of mind is somewhat vision-

ary or theoretical and not practical enough for the solution of this great problem. With all respect to the learned dean, I would point out that workers in the field of science are of two well-defined types, and this distinction holds true probably for all fields of human activity. In science, as in philosophy, there are minds that may be described as *minds of the first order.* These are the minds that blaze the way by seeking out new truths whether of science, philosophy, religion, or what not. At no one time in the history of the world has there been a large number of such minds as compared to the total population, and the entire list of such great prophets, poets, philosophers, and scientists is insignificant when one considers the vast number of people that have lived on this globe. It is true that this type of mind is usually visionary and impractical, for which Heaven be praised! but upon it rests all hope of future progress. The true scientist who is interested only in discovering new truths or the man whose mind is even strongly drawn in this direction is usually not capable of directing the energies of others. But the engineering type of mind (which may be described as belonging to the class of *minds of the second order*) is interested in applied science and is for the most part eminently practical when brought into contact with practical problems. It is a mind capable of high development along business and financial lines, while yet retaining in its background the powerful scientific methods of attacking problems which have made modern industry possible. And this is the great contribution that the engineer can bring to this problem of universal well-being. The one great thing we are all seeking is justice; but there is no justice where there is no knowledge, and the engineer, again using this term in its widest sense, alone possesses an accurate and intimate knowledge of industry.

No one would presume to say that the engineer alone can solve the industrial riddle, but it is clear that he can make a very great contribution to the solution. And it becomes increasingly clear that hope of a solution rests with the two groups just discussed. The National Department of Education has for some time been advocating strongly that engineering students be taught more of the fundamentals of business and that young men preparing for the field of business and commerce be instructed in some of the outstanding features of engineering. And it may be that we shall yet develop a combination of these two fields that will produce the new type of industrial leader. But whether the new leadership

comes from one field or the other, or from a combination of the two, the problem of universal well-being offers a challenge and a call to duty that the engineer may not refuse. The primary conditions and responsibility for universal well-being now rests upon his shoulders, and the engineer does not like to be concerned with a half-finished "job." There is good reason to believe he will do his share of work in attaining the ultimate results desired. Already there are significant signs that he has accepted this duty. The activities of city, state, and national engineering societies in all manner of civic problems in which they are competent to express an opinion need no rehearsal here. A few years ago it was unthinkable and unbelievable that an engineer should hold a high political office. Yet today we see with great satisfaction two engineers occupying gubernatorial chairs, one of them also an inventor of note, a Past-President of this Society and possessing a mind approaching at least that of the first order. More encouraging still we see an eminent engineer chosen, for the first time, to sit in the Presidential cabinet. These are pioneer adventures in a field that hitherto has been considered the exclusive territory of the lawyer and the politician, and their experiences are being watched with great interest by all thinking engineers. They are undoubtedly the vanguard of a larger invading army.

Whatever the result, we may be assured of one thing. Unless we can in some manner change our industrial system so that we can more nearly attain universal well-being and distribute the fruits of our industry more equitably, we have no reason for believing that our civilization shall endure, and its bones will full surely strew the shores of time along with those of the great civilizations that have preceded us. But if we can solve these problems, and I believe we can, there is hope that there may arise in America a civilization fairer than any that this war-weary world has yet seen, where universal well-being shall be a reality, and not a dream, and where liberty and justice shall prevail.

No. 1855

ANNUAL REPORT OF THE COUNCIL

1922

FOREWORD

THE Society's activities cover a field so broad that in its annual report the Council can outline only the major actions taken during the administration year of President Dexter S. Kimball. All of the organization activities are fully reported semi-monthly, in the A.S.M.E. News and in Mechanical Engineering, issued monthly.

The detailed reports of the Standing Committees of the Council have been published and were distributed at the Business Session of the 1922 Annual Meeting.

The 1922 Council was elected in the same way as last year, that is, through the regular Nominating Committee selected by the Local Sections and confirmed by the Society at the Annual Meeting, and this Committee so selected, prepared the ballot for officers of the Society in open session at the Spring Meeting.

The personnel of the Council for the year is given on pages vii and viii of this volume.

Seven regular meetings of the Council were held during 1922, the first on adjournment of the Annual Meeting of 1922 in December, the second March 6, at Kansas City, Mo., in connection with an engineering congress on the petroleum industry in which fifty-four organizations participated; two sessions in Atlanta in May during the Spring Meeting of the Society; New York and Springfield, Mass., September 25 and 26; October 21 at Ithaca, New York, and December 4, beginning the forty-third Annual Meeting of the Society.

The Council records with deep regret the death in office of Vice-President Louis Edward Strothman, and would place on permanent record the resolutions passed at the Atlanta Meeting:

> Whereas: in the death of Vice-President Louis Edward Strothman, the members of the Council and Society have lost a highly esteemed officer, and
>
> Whereas: in the death of Louis Edward Strothman the community has lost a citizen of great charm and endearing personality, who had won the affection of all with whom he came in contact:
>
> Resolved: that the Council of The American Society of Mechanical Engineers, in recognition of the qualities of its fellow member, and in token of its regret that such a career could not have been longer continued, for the benefit of the Society, the engineering profession and the public, by a rising vote, directs that this record be spread upon its minutes, and that a copy of these resolutions be sent to the family of Mr. Strothman.

By unanimous vote the place of Mr. Strothman was filled by Council election of Arthur L. Rice of Chicago. Mr. Rice has served the Society long and assiduously, both in the East and at Chicago, with a devotion vital to the Society's interests. He has been an active member of the Constitution and By-Laws, Local Sections and other major committees, and came to the Council with an intimate knowledge of the Society's affairs.

In August the Council appointed the Secretary, Calvin W. Rice, to represent the Society at the International Engineering Congress in Rio de Janeiro in connection with the Brazilian Centennial and International Exposition. Mr. Rice was also a delegate of ten other professional engineering organizations of the United States and Canada, from several civic organizations by President D'Oliver of the Sesqui-Centennial Exposition, organizing the celebration to take place in Philadelphia in 1926. The Hon. Herbert Hoover as Chairman of the International American High Commission appointed Mr. Rice to call upon each of the sections of the committee in South America. Major Fred J. Miller, Past-President, was appointed by the Council as Acting-Secretary during the absence of Secretary Rice.

A brief account of the Society's activities in the field of International Relations is included elsewhere in this report.

The Society has participated in an increasingly large number of functions and conferences. Representatives with the title of Honorary Vice-President were appointed to represent the Society at the following functions:

Electrical Safety Conference, John Price Jackson.

Advisory Board of the Bureau of Mines and Bureau of Standards, on Investigation of the Breakage and Heat Treatment of Rock Drill Steels, George T. Cousins.

National Rivers and Harbors Congress, Brig.-General Wm. H. Bixby.

DeLamater-Ericsson Tablet Committee, H. F. J. Porter, H. H. Suplee.

Commercial Engineering Conference, E. M. Herr, W. W. Nichols, W. H. Kenerson, R. L. Sackett, Walter Rautenstrauch.

Hobart College 100th Anniversary, Harte Cooke.

University of Southern California, Inauguration of Rufus Bernard von Kleinschmid, C. C. Thomas.

Virginia Agricultural and Mechanical College and Polytechnic Institute, Fiftieth Anniversary, J. S. A. Johnson.

Annual Meeting for the Industries of Pennsylvania, E. A. Fessenden.

American Association for Advancement of Science, Annual Meeting, Boston, Dugald C. Jackson, in addition to Dr. Hollis the representative of this Society on the A.A.A.S. Council.

U. S. Chamber of Commerce, Annual Meeting, O. P. Hood.

Pan-Pacific Commercial Conference, Honolulu, S. N. Castle.

American Construction Council, observers, H. L. Whittemore and George A. Weschler.

Department of Commerce, Informal Conference on Lumber Sizes, H. H. Snelling.

Seventy-fifth Anniversary of the Koninklijk Instituut van Ingenieurs, The Hague, Holland, R. A. Vander Willigen.

U. S. Chamber of Commerce, Committee on Fire Waste, H. A. Lacount, Ira N. Woolson.

Syracuse University, Inauguration of Chancellor Charles Wesley Flint, E. N. Trump.

Lehigh University, Inauguration of President Charles Russ Richards, Arthur W. Klein.

Informal Conference on Traffic Signals, E. W. Swartwout.

Washington Award of the Western Society of Engineers, Charles Russ Richards.

Twenty-fifth Anniversary of the Society of Mechanical Engineers of Japan, and the Thirtieth Anniversary of the Society of Naval Architects, Tokio, Japan, E. A. Sperry.

Engineering Museum, Representatives on Joint Committee, H. F. J. Porter and Reginald Pelham Bolton.

Special Committee Appointments on Society matters:

Tellers of Officers Ballot and Ballot on Amendments to the Constitution and Code of Ethics, W. H. Greul, Chairman, E. L. Sherwood and J. F. Callaghan.

During the year the Society's facilities at headquarters have been placed at the disposal of its own committees and the committees and governing boards of other organizations. Our records of 1922 show the following schedule: Administrative committees 41, professional committees 101, non-professional 24; committees of other societies, including American Engineering Standards Committee and the American Society of Safety Engineers 101, making a grand total of 267.

The Society has also continued to accommodate the American Society of Safety Engineers with temporary offices.

FINANCES

For some years the A.S.M.E. has enjoyed the position of having no indebtedness either capital or operating. Each year the Finance Committee prepares careful estimates of the income expected during the following twelve months and, coöperating with all committees having charge of activities, prepares a budget recommending expenditures not exceeding at the beginning of the fiscal year eighty-five per cent of the income. The remaining fifteen per cent is available for adjustments which have to be made during the year and for accumulating a small reserve. This latter has been necessary since the Society transacts business in volume of about $50,000 a month and because at times during the year disbursements have to be made before the income is received. A small margin has, therefore, been provided each year to build up a cash working capital. Several distinct groups of members serving at different times on the Finance Committee

have independently canvassed the situation and have reported that the minimum cash capital should not be less than one-third of the annual turnover. It will take several years to build up this reserve properly. Hereafter a minimum of fifty per cent of the initiation fees is to be immediately put in the Reserve Fund.

The average yearly amount paid in by a member in dues is between thirteen and fourteen dollars, whereas the corresponding gross expenditure per member is over thirty dollars.

The Auditor's Report, covering the fiscal year September 1921–October 1922, with detailed figures, is made part of the Report of the Finance Committee.

An important step was taken by the Council in the approval of a special committee report on solicitation of funds from the industries, and under the restrictions here outlined, the Council feels there can in no way be any reflection on the dignity of the Society. The Council finds a splendid precedent in the successful, and enthusiastic support of the industries in researches now being carried on in gears, steam tables, standardization of pipe fittings and fatigue of metals, partially financed also by the Engineering Foundation.

The Committee consisting of George A. Orrok, Chairman, W. C. L. Eglin, Arthur M. Greene, Jr., D. S. Jacobus, Fred J. Miller, C. T. Plunkett and W. L. Saunders, suggested the following restrictions which have been adopted by the Council as the policy in financing special researches and activities of the Society.

a Each suggested research must be presented on its individual merits, for approval by the Council that will in turn refer the matter to the appropriate authority or committee.

b Specific requests to the Council are to be accompanied with full details of proposed scope, method of solicitation of funds and budget.

c To favor and strongly urge the closest possible coöperation with universities and technical schools qualified and equipped to assist in the development and conduct of special research work.

d Coöperative, not competitive methods, to be worked out with existing research laboratories and activities in other organizations. Such coöperation could take the form of publication of papers and groups of papers where a definite industry desired to bring to the attention of engineers for the development of the industry, any problems of special research, without commercial bias.

e No exception is to be made to the Society being the custodian of all funds and having complete knowledge and control of the distribution and assignment of such funds through the Council, with the understanding always that no contributor is to be specially favored on account of any contribution for a research in which he is interested and that such contribution can be received only on the basis of general benefit to the industry.

MEETINGS

The Stated Meetings of the Society are the Annual New York meeting and the Spring Meeting, this year held in Atlanta, Georgia.

The visit to Atlanta was preceded by a visit to the University of Virginia, May 5 and 6, under the auspices of Local Sections and Student Branches. The Atlanta Meeting, May 8 to 11, included six strong Professional Sessions provided by the Professional Divisions, six miscellaneous papers presented at two general sessions and an important welding meeting. The self-supporting plan adopted at this meeting was a success. The splendid hospitality of the South was shown in the social events and visits to industrial centers.

To relieve the Society of the pressure at Annual and Spring Meetings and further, to provide important meetings for a larger proportion of membership, there has been developed through the initiative of the Local Sections and with the coöperation of the Meetings and Program Committee regional meetings, additional to the Spring and Annual Meetings. These regional meetings are organized by the Local Sections.

LOCAL SECTIONS

The Society has 57 Local Sections. These are listed in detail in the Year Book.

There have been coöperative meetings developed in the New York Sections of the A.S.M.E., A.I.M.E., A.I.E.E., and A.S.C.E.

Affiliation with the Engineering Society of Western Massachusetts has also been consummated.

President Kimball has been untiring during his year in his visits to the Sections and was able to get as far as the Pacific Coast. The Secretary also made several trips, individually, and with the President, and members of the Local Sections Committee have covered a wide territory. As the result of these visits to all sections of the United States, enthusiasm and interest have been developed in all parts of the country, particularly where local sections had not previously been so active.

A plan has been recently outlined and approved by the Council to care for publicity for papers before Local Sections.

PUBLICATIONS

The publications of the Society constitute an important agency in advancing its objects and nearly half of the Society's income is expended on them.

Two volumes of Transactions have been issued this year, Volume 42 and Volume 43. The preparation of these volumes has been carefully supervised by the Committee on Publication and Papers who endeavor to make the selection of papers cover what will be of value for permanent record.

Mechanical Engineering has twelve issues a year, which include technical papers, a survey of engineering progress, an engineering index, the monthly guide to current technical literature, the last later gathered into an annual volume The Engineering Index Annual. During the past fiscal year 850 pages of text have been published.

Condensed Catalogues of Mechanical Equipment this year has been changed to the larger and more impressive size, 9 by 12 inches, uniform with Mechanical Engineering and the leading publications in its field. It covers the product of 372 leading American manufacturers; also a Buyer's Guide of manufacturers of equipment — approximately 4200 firms are listed; 1000 engineers are classified under 400 different headings in the Directory of Consulting Engineers. Condensed Catalogues having now advanced to the position where it can carry free listing of all leading firms in certain departments, it is by that feature an inclusive publication of wide usefulness.

The Year Book for 1922 was brought out within this year, the issue for 1921 having been omitted.

The establishment of the A.S.M.E. News marked a step forward in the publication policy. It has been issued twice a month since its start with the December 22, 1921 issue. It was planned that this publication become a means of serving the membership and a better means for publishing news of the ever-increasing and so-called human activities of the Society. It supersedes the Society Affairs column of Mechanical Engineering.

MEMBERSHIP

The year's record of membership tells only a steadily increasing enrollment. Its deepest message is read in the obligations to united service on the part of members. The membership during the fiscal year has reached 16,696. The Changes in Membership are shown in Table 1.

Life Membership was awarded to Major Fred J. Miller by Special Council ballot. The presentation was made at the Business Session of the Annual Meeting with the following vote from the Council:

> The Council of The American Society of Mechanical Engineers desire to extend their thanks to Major Fred J. Miller for his recent service to the Society, and above all, to express their satisfaction in offering to him Life Membership in the Society, of which he has been for thirty-two years a member and supporter. They are conscious of his service as President and as a wise adviser and servant under all circumstances. Not the least of his service has been as Secretary of the Society, which he voluntarily assumed without pay, in order to permit Mr. Rice to represent the Engineers at the meeting in Rio de Janeiro.
>
> It is through such generous dedication of self that this Society has flourished and that will give it in the future a great influence upon the life of the Nation.

Other Life Membership awards are reported under the Awards and Prizes Committee.

The Council approved the policy of permitting such Local Sections as desire to comment on the qualifications of candidates from the respective Sections for admission provided comments are forwarded within twenty days. This will help the Membership Committee in its recommendations to the Council of desirable candidates, and should be of assistance in maintaining the highest standard of membership.

TABLE 1 CHANGES IN MEMBERSHIP

	Oct. 1, 1921	Oct. 1, 1922	Losses				Additions		Totals		
			Transferred from	Res'ig.	Dropped	Death	Transferred from	Elect.	Loss	Gain	Total
Honorary Members	19	19	...	...	...	1	...	1	1	1	....
Life Members	79	77	...	...	...	4	...	2	4	2	2
Members	7168	7733	...	42	66	65	140	597	173	737	564
Associate Members	3450	4023	90	22	30	5	152	568	147	720	573
Associates	744	844	11	3	10	3	7	120	27	127	190
Junior 15's	675	680	117	32	27	10	191	....	186	191	5
Junior 10's	2880	3265	272	47	18	5	...	728	343	728	385
Affiliates	.....	55	...	...	...	..	...	55	...	55	55
Totals	15015	16696	490	146	151	93[1]	490	2071	881	2561	1680

[1] The names of members and memorial notes appear in this volume.

PROFESSIONAL DIVISIONS

To encourage the group professional activities within the Society, the Council last year authorized Professional Divisions on

Aeronautics	Materials Handling
Forest Products	Ordnance
Fuels	Power
Gas Power	Railroads
Machine Shop	Textiles
Management	

Pending the organization of a Professional Division a Professional Group on Printing has been authorized and is in operation.

The Divisions enjoy complete autonomy. Each Division elects its own Executive Committee by letter ballot and the Chairmen of the Executive Committees of all the Divisions automatically constitute the Standing Committee on Professional Divisions, which in turn elects its own Chairman who has a seat on the Council. Members of the Society may enroll in any or all of the Divisions, but usually a member finds that his special interest is confined to one and not more than two Divisions.

The Divisions have with splendid enterprise assumed the task of providing papers for Annual and Spring Meetings of the Society, for the Regional Meetings, and for meetings of Local Sections. In addition the Management Division has sub-committees working on special subjects with one representative from the Society of Industrial Engineers, the Taylor Society, Investment Bankers Association and the National Association of Cost Accountants, such as Management Terminology, Standards for Graphics, Measurement of Managerial Ability. The Ordnance Division has devoted special attention to coöperation with the Ordnance Department of the U. S. Army concerning technical matters.

Other activities in research and standardization in the several subjects are contemplated as funds and interest develop.

CONSTITUTION AND BY-LAWS

The Committee on Constitution and By-Laws has continued most faithfully its work on the new Constitution, expressing the advanced concept of the Society's mission. The revised document simplifies terminology, and removes incidental inconsistencies. Fundamentals only are included, the administration details being relegated to By-Laws which can be adjusted also by Council without necessity of Society action.

The best authorities in the Society served as sub-committees of the Constitution and By-Laws Committee who have also given the phraseology the benefit of their many years of experience in the work of the Society. The Local Sections throughout the country have been particularly helpful, and in order that no mistakes be made, the attorneys of the Society and a well-known corporation lawyer have passed on the validity and completeness of the document.

The new Constitution went into effect with the announcement by the Presiding Officer at the 1922 Annual Meeting stating its adoption by letter ballot.

As a preliminary to the new Constitution, two amendments to the existing Constitution were carried during the year. The first extended the privilege of voting to Juniors, and the second simplified the manner of amending the Constitution.

RESEARCH, STANDARDS AND TECHNICAL COMMITTEES

The work of the professional committees covers a great variety of engineering subjects. For example, the Research Group is composed of thirteen committees; a second large group is that which at present is revising the A.S.M.E. Power Test Codes, of 1915. This group numbers 125 men and is sub-divided into 20 committees, a central or main committee of 25, and 19 individual committees of from five to thirteen specialists. Five of its revised codes are in final form.

The subject of dimensional standardization is each year receiving more and more of the Society's attention. At present 15 Sectional Committees are organized under the rules of procedure of the American Engineering Standards Committee for which the Society is sponsor and are at work on specific projects or are in process of organization.

The Society has one or more representatives on 13 additional Research Committees, Standards and Safety Codes.

The American Engineering Standards Committee is now in close touch with the National Standardizing bodies of thirteen foreign countries. The world-wide movement is still young but already there are indications that at least one international standard will soon be announced.

The Research and Standardization activity is likely to be considerably helped during the coming year with the inauguration of the policy of solicitation of funds from the industries recorded under "Finances" in this report.

BOILER CODE

The Boiler Code Committee comprises a body of twenty-three members with eleven sub-committees investigating various problems connected with the work. It is a continuing body and is constantly interpreting the Code to meet the application to various situations and the ever-changing conditions in the manufacture and operation of boilers and other pressure vessels.

The A.S.M.E. Boiler Code is operative as a legal construction code in the 17 States of

Arkansas	Minnesota	Oregon
California	Missouri	Pennsylvania
Delaware	New York	Rhode Island
Indiana	New Jersey	Utah
Maryland	Ohio	Wisconsin
Michigan	Oklahoma	

Cities: Detroit, Mich.; Erie, Philadelphia, Scranton and Allegheny County, Penn.; Kansas City, St. Louis, St. Joseph, Missouri; Los Angeles, Cal.; Chicago, Ill.; Nashville, Tenn.; Seattle, Wash.; Parkersburg, W. Va.; Omaha, Neb.; District of Columbia; Panama Canal Zone; Ontario, Canada; Shanghai, China, and the Hawaiian Islands have also adopted the Code.

It is also used as a standard by fifteen of the leading boiler manufacturers and by many of our foremost consulting engineers. The U. S. Government now specifies that boilers for many important Departments are to be constructed in accordance with the A.S.M.E. Boiler Code.

The Committee is now extending its activities to cover unfired pressure vessels, and during the year the Council has appointed members to serve on a special committee of the Boiler Code Committee to formulate this Code.

The Committee is planning to publish a new edition of the Code in 1923, and has ready the revision of the Heating Boiler Code.

EDUCATION AND TRAINING

The membership of the Student Branches totals 69 over a widely distributed area of colleges in the United States. The detailed list appears in the Year Book.

Student Branch members do not pay dues directly to the Society but are encouraged to subscribe to Mechanical Engineering at the special rate of two dollars per year, which is less than the actual production cost. The privilege of purchasing technical papers is extended to them on the same basis as members. Through this activity approximately 3,000 students are enrolled.

As an encouragement to student participation in Society affairs, two student prizes of twenty-five dollars each were established by Henry Hess, a member of the Council of the Society in 1915, to be awarded each year to members of Student Branches presenting the best two papers.

Dr. W. H. Kenerson, Chairman of the Committee on Relations with Colleges, has given most freely of his time and energy in visiting various colleges, since February of this year. His report is now in the hands of a special committee appointed by the President to take up certain suggestions of how best to secure the enthusiastic interest and coöperation of the great body of engineering students, to be helpful to them and they helpful to the Society. Opinions are now somewhat divided between maintaining Student Branches of the Society and encouraging Student Engineering Societies with representation for each of the national societies, and next year may see a coöperative plan worked out along these lines.

AWARDS AND PRIZES

Prof. Robert C. H. Heck was awarded life membership in the Society at the recommendation of the Committee on Awards and Prizes for the paper on Steam Formulas appearing in the 1920 Volume of Transactions.

Two Junior prizes were awarded this year. This prize was awarded for "the best paper or thesis submitted by Junior Members." These prizes went to R. H. Heilman for his paper on Heat Losses from Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Deg. Fahr., and to F. L. Kallam for a Preliminary Report on the Investigation of the Thermal Conductivity of Liquids. Mr. Heilman is an industrial fellow at the Mellon Institute in Pittsburgh, and Mr. Kallam is a graduate student at Leland Stanford Junior University.

Announcement of the awards was made at the Business Meeting of the Society on Wednesday, December 8. In regard to the Life Membership Award the Committee says:

"Invitations were sent to all of the members of the Society (over 16,000 in number) to send in a statement as to the opinion of individual members on the best paper which appeared in this volume of the TRANSACTIONS. It was a great disappointment that very few replies were received. The Committee feels in making such an award as this that its work will be greatly facilitated if it can have the hearty coöperation of the membership of the Society."

Columbia University in New York through its governing bodies, has extended to the A.S.M.E. the naming of the recipient of a scholarship in mechanical engineering.

The membership of the Committee on Awards and Prizes consists of Walter M. McFarland, Chairman; Ira N. Hollis, Spencer Miller, Henry B. Sargent and R. Sanford Riley.

JOINT ACTIVITIES

The Society shares in joint relation with the Founder Societies in the following interests:

Code of Ethics. The Code of Ethics voted on by the members of the Society resulted in a ballot of 3334 for, and 35 against adoption. This was the product of a report of a Joint Committee with representatives of four other national engineering societies. Our members of the Committee are A. G. Christie, Chairman, H. J. G. Hinchey, Chas. T. Main, J. V. Martenis, Robert Sibley. Professor Christie, Chairman of the Joint Committee, has labored for nearly ten years to achieve this result and received a special vote of thanks from the Council.

The John Fritz Medal for 1923 was awarded on July 6, 1922, to Senator Guglielmo Marconi, Honorary Member of the A.I.E.E., for his invention of Wireless Telegraphy.

Affiliated Societies. At the March meeting of the Society a petition from the American Society of Safety Engineers asking affiliation with the A.S.M.E. was received and granted.

The Engineering Foundation. This fund initiated at $500,000 in 1914 by Ambrose Swasey, Past President and Honorary Member of the A.S.M.E., it is hoped will be augumented soon by others. Alfred D. Flinn, also Secretary of United Engineering Society, has been appointed Director of

the Foundation. Mr. Flinn made a six weeks' tour to the Pacific Coast and intermediate points, with the object of interesting and securing the aid of engineers in a nation-wide plan for research relating to engineering and the industries. Among active projects are a research in fatigue phenomena of metals, investigation of arch dams, and the study of wood preserving processes, including paints and varnishes. The Foundation published annual reports and semi-monthly *Research Narratives.* During the year it also printed a Directory of Hydraulic Laboratories in the United States. It coöperated with National Research Council in establishing the Personnel Research Federation which publishes the *Journal of Personnel Research.*

National Research Council was established in 1910 under the auspices of the National Academy of Sciences with the coöperation of the engineering societies and Engineering Foundation. Through our connection with the Engineering Foundation we now have a permanent link between the engineering profession and the national organization of the men of science. The Engineering Division of the national Research Council includes representatives of fourteen national engineering societies. Its work has been financed in part by the Engineering Foundation. It has many active committees engaged in research projects connected with mechanical, electrical, welding, metallurgical, mining and civil engineering.

The term of Professor Greene on the Research Council expired this year, Albert Kingsbury was nominated and the appointment duly confirmed by the National Academy of Sciences. Professor Greene has contributed most effectively to advance and establish on a permanent basis the research work of the Society. He was Chairman of the Advisory Board of Mechanical Engineering Research connected with the Division of Engineering of National Research Council. He has been succeeded in this office by Professor Walter Rautenstrauch.

The F.A.E.S. and American Engineering Council. The F.A.E.S., the governing body of which is American Engineering Council, comprising 28 member organizations, in January, 1922, completed the first year of service.

To convey an idea as to the purpose, character and scope of the work of the F.A.E.S. a partial list follows of activities in which it has been engrossed through special committees, in many cases, giving untiring attention to the work in hand:

> Government Departments in Forms and Adjustments of Contracts; U. S. Civil Service in Classification and Compensation of Engineers; Department of Commerce, Elimination of Waste; Conference Boards in Jurisdictional Awards, Material Markets Conference, Bureau of Economic Research, Paving Bricks Standardization, Unemployment, Licensing of Engineers, and in Federal Legislation on Patents and Patent Office relief. Department of Public Works, Typographic Mapping, Reforestation, National Hydraulic Laboratory, Dyes Embargo, Kenyon Bill, having to do with the stimulation of industries and reduction of unemployment during period of cyclical depression. Revision of the Mining Laws; Research in Waste in Industry, the

Twelve-Hour Shift, Basic Wage Rates, and a study of the Muscle Shoals project.

International Engineering Congress of 1926. The Council has approved an International Engineering Congress in 1926, preferably in connection with the proposed Sesqui-Centennial in Philadelphia. An informal joint committee of representatives from the Founder Societies and the Engineers Club of Philadelphia has been appointed. Our representatives are L. P. Alford, representative of the President of the A.S.M.E., A. G. Christie, F. R. Low, I. E. Moultrop, D. Robert Yarnall, and Calvin W. Rice, Secretary. Representatives of the Society on the Board of Management are Governor James Hartness, Past-President, alternate H. P. Liversidge, and D. Robert Yarnall.

South American Trip of Secretary Calvin W. Rice. By appointment of the Executive Committee of the Council, the Secretary left for Rio de Janeiro on August 24, 1922, to represent by joint arrangement the principal Engineering Professional Societies of the United States and Canada at the International Engineering Congress, held in connection with the Centennial of Brazilian independence. He sailed on the *S. S. Pan-America* which also carried the U. S. Government official delegation to the exposition headed by the Secretary of State, Hon. Charles Evans Hughes.

In addition to our own Society, Secretary Rice bore credentials from

American Society of Civil Engineers
American Institute of Mining and Metallurgical Engineers
American Institute of Electrical Engineers
John Fritz Medal Board of Award
The Engineering Foundation
The United Engineering Society
The Engineering Division of National Research Council
The Federated American Engineering Societies
The Engineers' Club of New York
The Engineering Institute of Canada
The Engineers' Club of Philadelphia.

The Congress was formally opened on September 17, 1922, by Dr. Pires do Rio, Honorary President of the Congress and Minister of Public Works in Brazil. Sessions continued throughout October and the principal accomplishments of the Congress are indicated in the following resolutions:

PERMANENT ORGANIZATION OF THE INTERNATIONAL ENGINEERING CONGRESS

The first International Engineering Congress of the Americas, convened at Rio de Janeiro in September, 1922, is today and now declared to be a permanent organization which will devote itself to the establishment of more intimate relations with the engineers and industrial workers of the Americas for the purpose of furthering engineering and conferring an unbounded service on humanity.

The International Engineering Congress declares that the board of directors of this permanent organization consist of the presidents of each and every one of the scientific, technical and industrial organizations of the Americas, or a representative named by the president, once the directors of the organization and the president of the International Engineering Congress of the Americas have authorized the admission of his society into the body, and

That the general secretary be at once named who will occupy the post until the next meeting of the International Engineering Congress, or until a substitute shall be elected by vote of the presidents of the adherent societies, and

That the name of this permanent organization be "International Engineering Congress of the Americas," and,

That the secretary be authorized to communicate this resolution to the proper authorities through the proper channels, and

That at the same time he may be authorized to delegate his powers to competent persons whom he may elect to represent him when he cannot be present, and

That the general secretary elected, fully authorized to act, is Mr. Verne Leroy Havens.

STANDARDIZATION OF STATISTICS

Railroads and Ports

Standardization of Statistics for Customs, Ports and Railroads.

Considering that it is universally admitted that modern economic life cannot be carried on without well-kept, reliable statistics of customs, ports and railroads, and

Considering that it is also universally admitted that the acquisition, dissemination and interpretation of statistics pertains to government administration, and

Considering that it is a recognized fact that an extensive work of this kind should not be scattered but administered by a central organization,

The first International Engineering Congress approves the preliminary work that has been done in this regard by the International High Commission and respectfully asks the respective governments, through the International High Commission to draw up a general agreement on this subject, so that a measure which is so highly desirable may be accomplished without further delays.

Standardization of Methods and Technical Nomenclature

Considering that uniform definition of terms and specifications with regard to materials, whether natural resources or manufactured products, is the necessary basis for efficient production and is

likewise essential for the development of international commerce and industry, and

Considering also that in the engineering profession the preparation of nomenclature and specifications is the province of international congresses and of the national societies that take part in them,

It is resolved that the Congress appoint a permanent committee to study proposed definitions and specifications and present them to future sessions of the Congress to be approved and embodied as forms for international use.

The accomplishments of the Congress were actually tangible in that acting upon the recommendation, the Government of Brazil requested the Government of the United States to loan an officer of the Bureau of Standards to introduce the methods proposed by Mr. Hoover and transmitted through Mr. Rice. Further, there was a development by a group of engineers from the United States in Rio de Janeiro, under chairmanship of Col. C. H. Crawford, member of A.S.M.E., of a list of technical words and phrases in the Portuguese language, and it is possible that a similar list of Spanish technical words and phrases will later be prepared by other centers visited by the Secretary.

Through introductions furnished by the Secretary of Commerce, Hon. Herbert Hoover, acting as Chairman of the Inter-American High Commission and by Dr. L. S. Rowe, Director-General of the Pan-American Union, the Secretary continued his trip through the other countries of South America, thus developing further engineering society affiliations, visiting the cities of Montevideo, Buenos Aires, Santiago, Lima, San José and Havana.

In spite of the brevity of his visit, Mr. Rice was convinced by the "teamwork" that was done that engineers in South America are eager to work with engineers of other countries in service to the profession and to the public. In each of these centers he urged an amplification of the domestic policies of the Society, namely —

1. That every professional man from the United States affiliate with the local engineering society of the country where he is residing, even if only temporarily
2. That they form within or under the auspices of the local society groups of members of the National Engineering Societies of the United States for purposes of service to the local society
3. That the local society link up with its government
4. That the organizations of engineers — members of U. S. societies link up —

   a With their country's government through the embassy and commercial attaché and with the Inter-American High Commission

   b With the National Engineering Societies of the United States.

Throughout his entire trip the Secretary reported a lively interest in engineering and in organizations of engineers. A complete account of this

trip was published in MECHANICAL ENGINEERING in January, 1923, and in the A.S.M.E. NEWS.

EMPLOYMENT SERVICE

During the first part of the year the Employment Bureau Service was turned over to the F.A.E.S., but after a short term turned back to the management of the Founder Societies of the United Engineering Society.

The number of men registered and placed is indicated below:

	Jan	Feb	Mar	Apr	May	June	Jul	Aug	Sept	Oct	Nov	Dec	Total
No. of men reg'd.	142	138	141	122	152	152	89	100	83	121	138	98	1,477
Total men placed	131	114	170	218	263	315	209	263	250	252	226	193	2,604

The total 1,477 refers only to the number of men registered during the year, the number of names on file being from 3,000 to 4,000.

A joint committee has been organized that is to report on the possibility of making this service self-supporting. Meanwhile, the service is being conducted by the Secretaries of the four societies as it was before it came under the F.A.E.S. A more detailed report is made a part of the Standing Committee Reports.

LIBRARY

Some statistics of the Engineering Societies Library not given in the annual report of our committee will be of interest:

During 1922 a total of 3,353 volumes (1,769 by gift, 1,584 by purchase), 616 pamphlets (601 by gift, 15 by purchase), and 25 maps and plans, making a total of 155,207 now in the permanent collection.

Expenditures for books, periodicals, supplies, and salaries were approximately $30,000. In addition about 8,000 non-visitors were assisted by mail or telephone. The average daily attendance was 86.

The Service Bureau made 226 searches and translated 103 articles totaling 343,130 words. It prepared 22,087 photoprints for 2,421 persons. The receipts for this work were about $14,500.

The task of recataloging the collection still occupies the chief attention of the Library Staff. During the year 16,678 volumes were recataloged and 149,980 cards prepared and added to the catalog. The catalog now contains specific references to 50,312 subjects.

Special attention has been given to reduce the accumulation of unbound material in the Library and more than 3,000 volumes were bound during the year.

The Carnegie Corporation and the National Electric Light Association have continued their contributions to the maintenance of the Library and the Society of Naval Architects and Marine Engineers has also become a contributor.

No. 1856

THE SIZE OF DRY-VACUUM PUMP TO EMPLOY IN A GIVEN CASE

A RAPID AND PRACTICALLY ACCURATE METHOD FOR ITS DETERMINATION, APPLYING TO ANY SET OF CONDITIONS

By Edward W. Noyes [1]

and

Harold V. Sturtevant [1]

Non-Members

In pumping from atmospheric pressure down to a given vacuum with a dry-vacuum pump, as during the process of exhausting a closed tank, the volumetric efficiency of the pump varies from that at atmospheric pressure to that corresponding to the given vacuum and an average value has to be determined to use in the theoretical formulas which apply to this kind of service.

Two methods of determining the average volumetric efficiency were successively tried out by the authors and the results plotted in the form of curves, but these results did not check satisfactorily with those obtained in actual tests of an installation. The process of calculation employed in the second (and somewhat more accurate) method was then reversed, and, working with actual test data, values were obtained from which a "constant" curve was plotted, this curve giving closely accurate results from 80 per cent perfect vacuum up. With the aid of this "constant" curve two charts have been plotted, by means of which it is possible to determine rapidly and with sufficient accuracy (1) *the size of pump required to exhaust a given volume to a specified degree of vacuum in a predetermined time; or* (2) *where a vacuum pump is already installed, the time required to exhaust a given volume to a specified degree of vacuum.*

DRY-VACUUM PUMPS are subjected to two different kinds of service, one of which may be classified as continuous vacuum service, in which the intake and delivery pressures remain practically constant during the operation of the pump; and the other as variable vacuum service, in which the intake pressure varies

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from atmosphere down to some low vacuum during the operation of the pump, as would be the case in exhausting a closed tank.

2 The dry-vacuum pump is employed on the latter class of work in many of the industries, notably in the following:

a Creosoting and Impregnation of Wood
b Dehydrating of Food Products
c In Textile and Dye Works to Assist in the Penetration of the Fabrics with Dyes
d In Electrical Establishments, to Remove Air from Insulation
e In the Extraction of Wax from Waste Wax Paper.

3 The problem of determining the size of vacuum pump required to evacuate a tank of given size, to a certain degree of vacuum, in a predetermined time seems to be a rather elusive one, and the purpose of this paper is to present a method for its solution which will apply to any set of conditions, and which may be quickly used to determine the correct size of pump, with accuracy sufficient for all practical purposes.

4 The experimental work described was conducted at the Claremont, N. H., plant of the Sullivan Machinery Company, and the machines used were the standard, straight-line, belt-driven vacuum pumps manufactured by that company.

METHOD AND PROCEDURE

5 Since it became necessary to make use of the characteristic volumetric-efficiency curve of a vacuum pump, which holds true in the continuous type of service, a brief review of the method used to obtain this curve will be given. This method is the one presented by S. B. Redfield at the Spring Meeting of The American Society of Mechanical Engineers in May, 1921, in a paper entitled Capacity Tests of Dry-Vacuum Pumps by the Low-Pressure Nozzle Method, Fig. 1 showing the arrangement of the apparatus. The following is quoted from that paper:

> In Fig. 1 the vacuum pump is drawing from tank *A*, and any desired vacuum may be produced in *A* and the pipe connecting it to the pump by adjusting throttle *B*. Throttle *B* is a globe valve, provided with a long handle for close adjustment. The running vacuum is observed on mercury column *E*. Tank *C* has at its end the nozzle *D*, taking air from the atmosphere, and the pressure drop through *D* is shown by the water column *F*. A thermometer *G* shows the temperature of intake to vacuum-pump cylinder and thermometer *H* shows the temperature of the air entering the nozzle.

We now see that the apparatus is composed of three parts: the pump to be tested; tank A and piping in which any desired vacuum is to be maintained, measured by mercury column E and at temperature of thermometer G; and tank C, which by means of nozzle D, water column F and thermometer H, measures the amount of air handled. As B is adjusted, the quantity of air handled will change. This will change the vacuum as shown on the gage E, and water column F will also change to measure the new quantity of air passing through the nozzle D. As the drop through D is small, the pressure in C is well above the critical at all times and the quantity of air passed will change with the slight vacuum in C, as measured by water column F. Thus, there is a wide range of air quantities which can be made to pass through a low-pressure nozzle of a given size, and by proper adjustment of throttle B any number of test points may be obtained as close together or as far apart as desired, independently

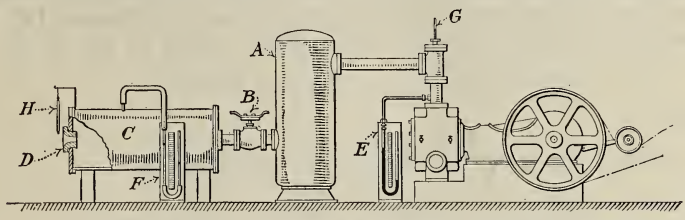

Fig. 1 Apparatus for Low-Pressure Nozzle Test of Vacuum Pumps

of the speed of the pump. Furthermore, any specified vacuum may be secured at will, up to the limit of the machine.

6 The formula mentioned in Mr. Redfield's paper was used in the computations of the characteristic volumetric efficiencies. It is:

$$Q = 3.64\, Cd^2 \sqrt{\frac{HT}{P_m}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where —

Q = cubic feet flowing per minute at absolute pressure of downstream side of nozzle and temperature shown by thermometer on upstream side of nozzle

C = coefficient of flow = 0.97 to 0.99

d = diameter of nozzle throat (smallest section), in.

H = observed nozzle water column, in.

T = absolute temperature (deg. fahr.) shown by thermometer on upstream side of nozzle

P_m = absolute mean pressure; that is, the average between the pressures on upstream and downstream sides of nozzle, lb. per sq. in.

7 In the following computation the temperature at thermometer G was so nearly equal to the temperature at thermometer H that it was assumed to be the same.

8 A sample calculation of the volumetric efficiency was made from a test on a 14-in. by 6-in. pump, as follows:

Observed Data:

Vacuum, inches of mercury	15.00
Water column, inches	2.47
Room temperature, deg. fahr	64
Barometer, inches of mercury	28.96
R.p.m. of pump	301
Diameter of nozzle, inches	2.0
Atmospheric pressure, lb. per sq. in. (= 28.96 × 0.4912)	14.22

Pressure on the downstream side of the nozzle is equal to the atmospheric pressure minus the pressure shown by the water gage. The average between the upstream and the downstream pressures will give the P_m for Formula [1]. Thus the downstream pressure = 14.22 − (2.47 × 0.036) = 14.12 lb. per sq. in.

$$P_m = \frac{14.22 + 14.12}{2} = 14.170 \text{ lb. per sq. in.}$$

Therefore —

$$Q = 3.64 \times 0.98 \times 2^2 \sqrt{\frac{2.47 \times (460 + 64)}{14.170}} = 136.43 \text{ cu. ft.}$$

Cubic feet of free air flowing per minute —

$$= 136.43 \times \frac{14.12}{14.22} = 135.57$$

Delivery at the vacuum shown by the mercury gage —

$$= 135.57 \times \frac{28.96}{13.96} = 280.63 \text{ cu. ft.}$$

Piston displacement (P.D.) per minute:

P.D. per revolution = 1.066 cu. ft.

P.D. per minute = 1.066 × 301 = 321 cu. ft.

Volumetric efficiency at intake conditions —

$$= \frac{280.63}{321.00} \times 100 = 87.4 \text{ per cent.}$$

9 This procedure was followed for each inch of mercury from 5 in. up to closed suction and the curve shown in Fig. 2 determined. This curve is an average of a number of curves made from tests on pumps of different sizes, including 14 in. by 6 in., 18 in. by 8 in., 22 in. by 10 in. and a duplex 21 in. by 12 in. The average curve checks very closely with all the individual curves and it may be taken as representative for all dry-vacuum pumps of this general type. That is, up to 50 per cent of perfect vacuum the volumetric efficiency is above 90 per cent. The "critical" vacuum is about 75 per cent, at which point the volumetric efficiency is 78 per cent.

10 In pumping from atmospheric pressure down to any given vacuum, as during the exhausting of a closed tank, the volumetric efficiency varies from that at atmospheric pressure to that cor-

responding to the given vacuum and an average value has to be determined to use in the formula which applies in this kind of service, which from Harris' Compressed Air is:

$$v = \frac{V}{\left(\frac{P_n}{P_a}\right)^{\frac{1}{n}}} - V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

or

$$n = \frac{\log P_a - \log P_n}{\log (V + v) - \log V} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where —

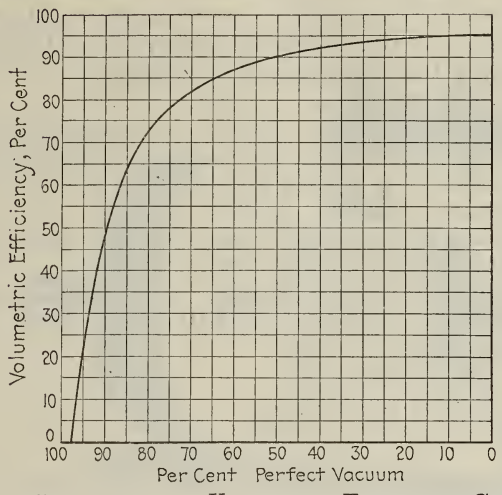

FIG. 2 CHARACTERISTIC VOLUMETRIC-EFFICIENCY CURVE OF DRY-VACUUM PUMPS

v = theoretical piston displacement in cubic feet per stroke
V = volume of tank to be exhausted, cubic feet.
P_a = atmospheric pressure, inches of mercury
P_n = pressure in tank after n strokes, inches of mercury
n = number of strokes of pump required to produce required vacuum in tank.

11 Isothermal conditions are assumed in the derivation of this formula. This is not actually true, but it is close enough for all practical purposes.

12 Formulas [2] and [3] give theoretical values for v and n based on 100 per cent volumetric efficiency, and account must be taken of the average volumetric efficiency under which the pump

operates in order to determine the actual piston displacement or revolutions required. Because of the fact that the vacuum changes at a very rapid rate during the first part of the exhausting process and at a very slow rate during the latter part, the average volumetric efficiency under which the pump operates in pumping down to any given vacuum cannot be determined by dividing the total area under the characteristic volumetric-efficiency curve by the length down to the given vacuum, as would be the case if the vacuum varied at a uniform rate.

13 The average volumetric efficiency determined in this manner gives results which do not check up with those obtained from an actual installation. The following example is a typical case and will serve to show the inaccuracy of the method.

With an 18-in. by 8-in. pump, 1.17 cu. ft. piston displacement per stroke, it takes 545 revolutions to exhaust a volume of 245.211 cu. ft. from atmospheric pressure to a vacuum of 27 in. with a barometer of 28.77 in.

Per cent of perfect vacuum = 27/28.77 = 93.8

Area under the characteristic volumetric-efficiency curve up to 93.8 per cent = 19.82 sq. in. Length = 4.663 in.

Average height = 19.82/4.663 = 4.25 in. Height for 100 per cent = 5.00 in.

Average volumetric efficiency = 4.25/5.00 = 85.0 per cent

$$n = \frac{\log P_a - \log P_n}{\log (V + v) - \log V}$$

$$= \frac{\log 14.12 - \log 0.870}{\log (245.211 + 1.17) - \log 245.211}$$

$$= 585.46$$

This is the total number of strokes required with a pump of 100 per cent volumetric efficiency.

14 The actual total revolutions required by this method $= 585.46/(2 \times 0.85) = 344.5$. Since the revolutions required in an actual test were 545, this method will give a result which is 37 per cent under the real requirements. The fallacy of this method lies in the fact that the degree of vacuum is assumed to vary directly with the time of operation. If one watches the mercury gage during the exhausting of a closed tank, it will be seen that as the pump starts up the mercury column begins to rise very rapidly, but it gradually slows down, until when it is up around 25 to 26 in. it is moving very slowly. From this it is evident that the number of revolutions required to obtain each succeeding inch of mercury will be correspondingly greater than the number required to reach the preceding inch.

METHOD USED FOR ACCURATELY DETERMINING THE VOLUMETRIC EFFICIENCY

15 From the foregoing discussion it is very evident that for accurately determining the average volumetric efficiency, some means must be used to record the number of revolutions made by the pump during definite intervals of time throughout the total interval taken to reach the desired vacuum. Since it is more convenient to measure revolutions, a recording counter was devised for this purpose. This counter is shown in Fig. 3. It consists of a wooden drum D upon

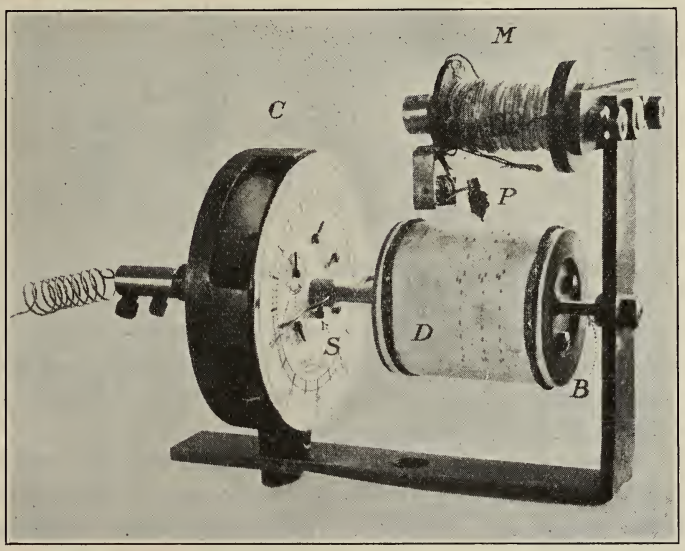

Fig. 3 Recording Revolution Counter Used in Tests

which is wound a roll of paper and which is direct-connected to the dial hand of an ordinary 100:1 revolution counter C. Above the drum is mounted a pencil P and an electromagnet M, the latter being so arranged and connected that a flow of current through the coil will cause the pencil to mark the paper. The electromagnet is operated by a push button so located at the mercury gage that as the column passes each inch graduation on a scale the button is pressed and a mark made on the paper. After the test is finished the hand of the counter is loosened from its shaft by loosening the set screw S, the drum hand turned back to the starting point, and the number of revolutions at each mark recorded. The drum is rotated on a threaded shaft B, so that it travels along the shaft

each revolution and thus avoids confusion of the marks. By means of this apparatus the total revolutions at each inch of mercury is obtained.

16 The method employed in computing average volumetric efficiency is illustrated in the following table, the successive steps being shown by columns (1) to (8).

(1) Vacuum in Inches of Mercury	(2) Total Rev. per Counter	(3) Δ Rev.	(4) Vol. Effy. (V.E.) from Characteristic Curve	(5) $\frac{1}{2}\Delta$ V.E. + V.E.	(6) Col. (3) × Col. (5)	(7) Summation of Col. (6)	(8) True Avg. Vol. Effy. Col. (7) ÷ Col. (2)
5	18.5	18.5	94.5	94.8	1754.0	1754.0	94.8
6	22.0	3.5	94.3	94.4	330.5	2084.5	94.7
7	26.5	4.5	94.0	94.2	423.0	2507.5	94.6
8	32.0	5.5	93.7	93.9	516.0	3023.5	94.5
9	37.5	5.5	93.3	93.5	514.0	3537.5	94.3

17 Following this same procedure the average volumetric-efficiency curve was calculated for several different sizes of pumps working with variable volumes and also under different barometers. The final average volumetric-efficiency curve was taken as an average of all these curves. This curve is shown in Fig. 4. Note that the stretch from 25 in. to 26 in. is "critical."

18 It was expected that the use of this average volumetric efficiency with the formula given would make possible the determination of the proper size of pump to use in any case. For example, if the size of tank to be evacuated, the total number of revolutions allowable, and the atmospheric and final pressures were known, then the theoretical piston displacement required per stroke could be calculated. Dividing this by the average volumetric efficiency corresponding to the given vacuum would supposedly give the actual piston displacement required per stroke. Or on the other hand, if the volume to be evacuated, the piston displacement per stroke, and the atmospheric and final pressures were known, then the theoretical number of revolutions required to produce the given vacuum could be calculated. Dividing this by the average volumetric efficiency corresponding to the given vacuum would supposedly give the actual number of revolutions required to produce the given vacuum. When this method was applied to the results taken from actual tests, however, it was found that the two did not check out

so closely as desired, and it consequently became necessary to attack the problem of finding the average volumetric efficiency in another way.

DETERMINATION OF CONSTANT FOR USE WITH GENERAL FORMULA

19 In doing this the process was reversed and a new set of values determined for the average volumetric efficiency by first solving the formula for v and then dividing this result by the actual piston displacement per stroke of the pump used. This same process was worked through for a number of cases and the average of the

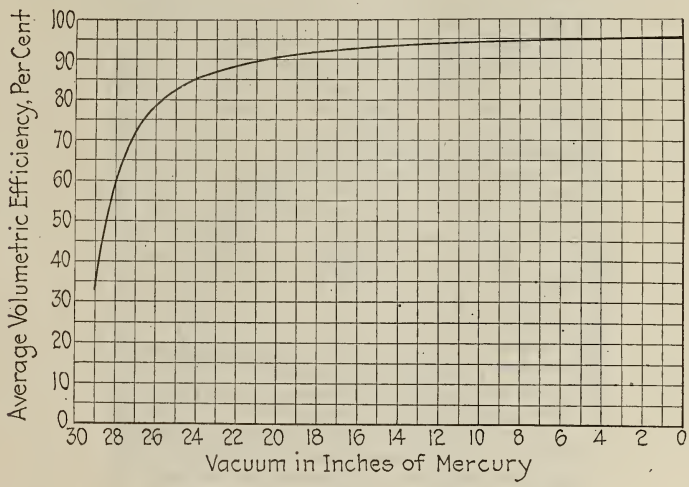

FIG. 4 AVERAGE VOLUMETRIC EFFICIENCY OF DRY-VACUUM PUMP DOWN TO ANY VACUUM WHEN EXHAUSTING A CLOSED SPACE TO THAT VACUUM

curves obtained was plotted and termed a "constant curve" for use with the general formula. A sample calculation of one of the points on this curve follows.

Pump, 18 in. by 8 in.; piston displacement, 1.17 cu. ft. per stroke; vacuum, 24 in. with barometer of 28.77 in., or 83.5 per cent perfect vacuum; total revolutions = 250, or n = 500; volume evacuated = 245.211 cu. ft.
Then —

$$P_a = 28.77; \qquad P_n = 28.77 - 24.00 = 4.77$$

$$v = \frac{245.211}{\left(\frac{4.77}{28.77}\right)^{0.002}} - 245.211$$

$$= 0.889 \text{ cu. ft.}$$

The actual piston displacement of pump, however, is 1.17 cu. ft. whence —

$$\text{Constant} = \frac{0.889}{1.170} = 0.76 = K$$

20 Fig. 5 shows this constant curve plotted against percentage of perfect vacuum. The individual constant curves of which Fig. 5 is an average show some divergence between zero and 80 per cent perfect vacuum, so that within this range the results obtained by using the chart are slightly inaccurate. From 80 per cent perfect vacuum up, however, all the individual curves practically coincide and the values given by the chart are accurate for all practical purposes. Fig. 6 enables a direct comparison to be made between the average volumetric-efficiency curve and the constant curve.

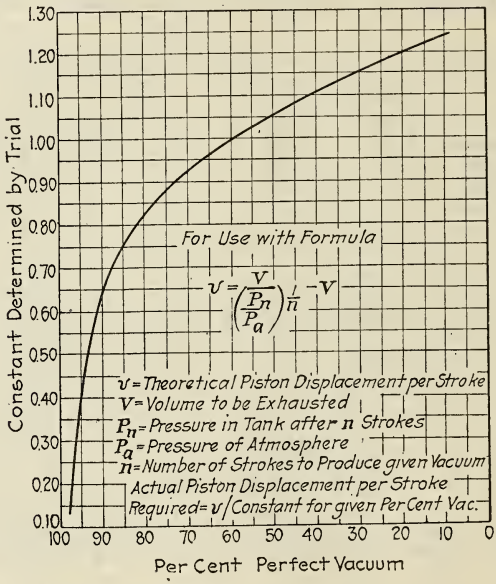

Fig. 5 Vacuum-Pump Constant for Use with General Formula for Piston Displacement

The reason why the two curves do not coincide is probably due to the unavoidable leakage entering into the tests. In order to obtain the average volumetric-efficiency chart it was necessary to make use of the characteristic volumetric-efficiency curve which was obtained under conditions where the leakages had their full effect. Under the conditions prevailing during the exhaustion of a closed tank the effects of the leakage are minimized during the first part of the process and exaggerated during the latter part. The reasons for this are: during the first part of the exhaustion the pressure differences are very slight and the time during which these leaks are exposed is very short; on the other hand, during the latter part of

the process the pressure differences are very great and the time of exposure is very long. Thus at the lower vacuums around six and seven inches of mercury the constant curve rises to values higher than those on the average volumetric-efficiency curve; the curves are very close together around 20 in. and 22 in., showing that at this point the leakages in both tests were practically the same; at the higher vacuums the constant curve falls under the average curve, due to the fact that the leakage has great effect at this point.

21 From the results of the preceding work it is evident that

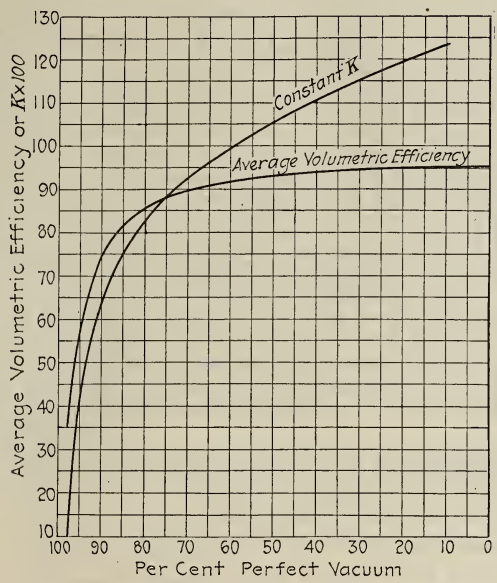

FIG. 6 COMPARISON OF AVERAGE VOLUMETRIC EFFICIENCY AND CONSTANT DETERMINED BY TRIAL

the curve to be used in connection with the general formula for solving any vacuum-pump problem is the constant curve rather than the average volumetric-efficiency curve, because the former has been determined from the results of tests on an actual installation.

22 In the general formulas [2] and [3] there exist direct and inverse relationships between the variables which may be taken advantage of in plotting the curves. For example, in Formula [2], v varies approximately inversely as n and directly as V, and in [3], n varies approximately directly as V and inversely as v.

23 This relationship makes it possible to plot Formula [2] for some given values of V and n, such as $V = 1000$ and $n = 600$,

and with v and the percentage of perfect vacuum as the variables. Then for any given case where the values of V and n are different from those in the plot, the value of v required would be directly proportional to V and inversely proportional to n. Likewise Formula [3] can be plotted for some given values of V and v, such as $V = 100$ and $v = 0.5$, and with n and the percentage of perfect vacuum as the variables. Then for any given case the value of n required would be directly proportional to V and inversely proportional to v.

24 The value of v thus obtained was multiplied by 2 and

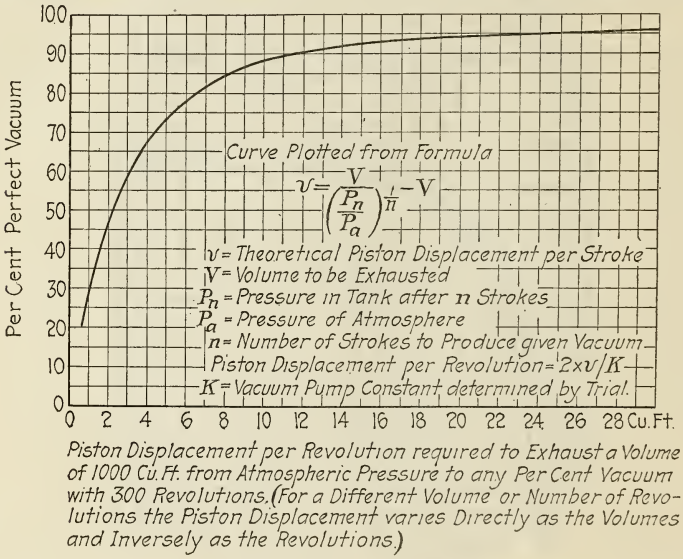

FIG. 7 DRY-VACUUM PUMP PISTON DISPLACEMENT REQUIRED TO EXHAUST A CLOSED SPACE

divided by the constant determined by trial and plotted as piston displacement per revolution in Fig. 7. The value of n was divided by 2 and by the constant and plotted in Fig. 8 as total revolutions required.

CHARTS FOR USE IN CALCULATIONS

25 Fig. 8 will apply to cases where the displacement of the pump is assumed or known and it is desired to find the total number of revolutions required to exhaust a given volume to a predetermined vacuum. Knowing the time which may be used in the process, the r.p.m. of the pump may be obtained. A barometric pressure of 29.5 in. was used in the computations and the results

plotted against percentage of perfect vacuum, so that the curve will apply with any barometric pressure. In using the curves, the average barometric pressure over a long period of time in any given locality should be used. Knowing the degree of vacuum desired, the percentage of perfect vacuum can be readily calculated. Fol-

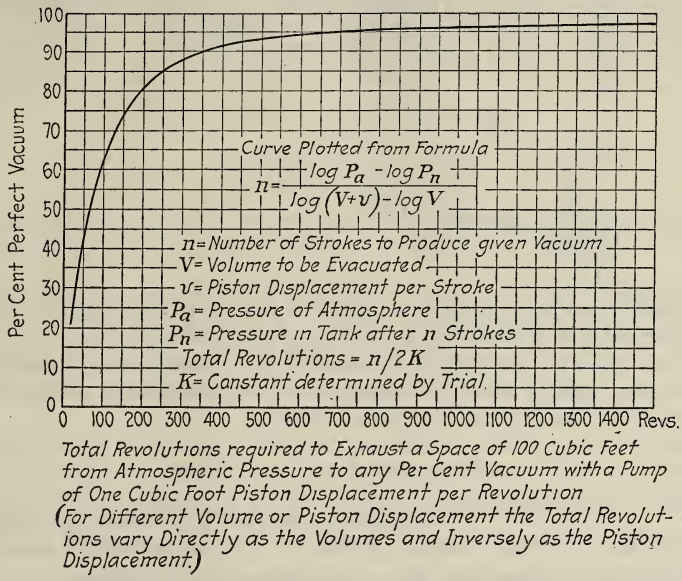

FIG. 8 DRY-VACUUM PUMP REVOLUTIONS REQUIRED TO EXHAUST A CLOSED SPACE

lowing is a sample calculation used in the determination of this curve.

Barometer = 29.5 in.; vacuum in inches of mercury = 24; percentage of perfect vacuum = (24/29.5) × 100 = 81.3; V = 100 cu. ft.; v = 0.5 cu. ft. Then —

$$P_n = (29.5 - 24) \times 0.4912 = 2.70 \text{ lb. per sq. in.}$$
$$P_a = 29.5 \times 0.4912 = 14.49 \text{ lb. per sq. in.}$$

Also, from Fig. 5,

$$K = 0.81$$

Whence —

$$\text{Total revolutions} = \left(\frac{1}{2} \times \frac{\log 14.49 - \log 2.70}{\log (100 + 0.5) - \log 100}\right) \div 0.81$$
$$= 207.95.$$

Hence at 81.3 per cent perfect vacuum is plotted 207.95 revolutions. This same process was followed for about twenty different points and the curve thus obtained.

26 Fig. 7 will apply to problems in which the total revolutions or r.p.m. and time are known or assumed, and it is desired to calculate the piston displacement necessary to exhaust a given volume to a predetermined vacuum. A sample calculation used in the determination of this curve follows.

Barometer = 29.5 in.; vacuum in inches of mercury = 25; percentage of perfect vacuum = (25/29.5) × 100 = 84.7; V = 1000 cu. ft. Then —

$$P_n = (29.5 - 25) = 4.5 \text{ in.} \qquad P_a = 29.5 \text{ in.}$$

$$n = 600$$

$$v = \frac{V}{\left(\frac{P_n}{P_a}\right)^{1/n}} - V$$

$$= \frac{1000}{\left(\frac{4.5}{29.50}\right)^{0.0016667}} - 1000$$

$$= 3.138$$

Actual piston displacement per revolution = 3.138 × 2/0.76 = 8.258 cu. ft., where 0.76 is the value of K from Fig. 5 for 84.7 per cent perfect vacuum.

Thus piston displacement of 8.258 cu. ft. per revolution was plotted against 84.7 per cent perfect vacuum. This process was repeated for many different percentages of vacuum and the curve obtained.

ILLUSTRATIVE CALCULATIONS

27 The problems which may be solved with the aid of Formulas [2] and [3] will vary in nature, but a few typical cases will be worked out in order to illustrate the application of the curves.

28 First and most important is the case where it is desired to determine the size of pump required to exhaust a given volume to a certain degree of vacuum in a predetermined time. This problem may be approached from two different angles: the r.p.m. of the pump may be assumed and the required piston displacement per revolution determined, or the piston displacement per revolution may be assumed and the total revolutions or r.p.m. calculated.

I — Given a volume of 5000 cu. ft., including the piping between the pump and tank, it is desired to find the size of pump required to exhaust the tank to a vacuum of 26 in. in 30 min. Barometer, 29.5 in.

a Assume that the pump will run at 280 r.p.m. In this case the percentage of perfect vacuum = 26/29.5 = 88.1. Using chart of Fig. 7, follow line at 88.1 per cent horizontally until it intersects the curve, then vertically downward to the base line, when the piston displacement will be found to be 10.15 cu. ft. This is the piston displacement per revolution required to exhaust 1000 cu. ft from atmospheric pressure to 26 in. with 300 revolutions of the pump. Hence —

$$\text{Displacement per revolution} = \frac{10.15 \times 5000}{1000} \times \frac{300}{280 \times 30} = 1.814 \text{ cu. ft.}$$

This is the actual piston displacement required with a pump running at 280 r.p.m. to exhaust 5000 cu. ft. to 26 in. in 30 min. Allowance has been made for leaks such as are always present in the average installation. For excessive leakage, however, a small factor of safety may be added.

b Assume piston displacement per revolution = 2 cu. ft. Using chart of Fig. 8, follow line at 88.1 per cent horizontally until it intersects the curve; then vertically downward to the base line, where the number of revolutions is found to be 305. This is the total number of revolutions required to exhaust a volume of 100 cu. ft. from atmospheric pressure to 26 in. with a pump of 1 cu. ft. displacement per revolution. Hence —

$$\text{Total revolutions} = \frac{5000}{100} \times \frac{305}{2} = 7625$$

$$\text{R.p.m.} = \frac{\text{Total revolutions}}{\text{Time in minutes}} = \frac{7625}{30} = 254$$

This is the actual r.p.m. required with a pump of 2 cu. ft. piston displacement per revolution to exhaust 5000 cu. ft. to 26 in. in 30 min.

29 The second type of problem, the solution of which may be obtained by use of the curves, is the one in which a vacuum pump is already installed and it is desired to determine the time required to exhaust a given volume to a certain degree of vacuum.

II — Given an 18-in. by 8-in. pump running at 300 r.p.m., how long will it take to exhaust a tank of 2500 cu. ft. capacity from atmospheric pressure to 26 in. vacuum? Barometer, 28.4 in. Piston displacement per rev. = 2.34 cu. ft. percentage of perfect vacuum = 26/28.4 = 91.5.

Using chart of Fig. 8, follow line of 91.5 per cent horizontally until it intersects the curve, then vertically downward to the base line, when the revolutions are found to be 415. This is the total number of revolutions required to exhaust a volume of 100 cu. ft. from atmospheric pressure to 26 in. with a pump of 1 cu. ft. piston displacement per revolution. Hence —

$$\text{Total revolutions} = \frac{2500 \times 415}{100 \times 2.34} = 4433.8$$

$$\text{Time required} = \frac{\text{Total revolutions}}{\text{r.p.m.}} = \frac{4433.8}{300} = 14.8 \text{ min.}$$

This is the actual time required, allowance having been made for leaks such as are present in the average installation.

CONCLUSION

30 In presenting this method of solving problems concerning the evacuation of closed spaces, the authors do not claim to have produced an absolutely accurate rule which will answer in every case. They do hold, however, that the method proposed of using a constant determined by trial gives much more accurate results than can be obtained by the use of an average volumetric efficiency computed by either of the two methods mentioned.

31 It will be noticed that the question of the effect of varying amounts of leakage on the size of pumps required has not been considered. This is undoubtedly an important phase of the problem and in any accurate determination of the size of pump or time required for evacuation, as the case may be, the leakage area of the system should be previously determined.

32 Obviously, if the system under consideration has two or three times the leakage area that was present in the system from which the curves of this paper were obtained, then the size of pump determined from them will be too small, or the number of revolutions required to exhaust the system in the given time will be too small. Yet even had this leakage factor been included, it is doubtful whether in the majority of cases it could be made use of.

33 Take the case of a purchase of a vacuum pump for evacuating a closed system. The chances are that the buyer has no means at hand for determining the leakage area before the pump ordered arrives, and it is unlikely that he would first install the closed system and have a test made on it before determining the size of pump required. This of course applies to an average commercial installation where the buyer, if he specifies a ten-minute period that he wishes to allow for the evacuation, will not care much if the pump provided actually requires, say, eight or twelve minutes in which to do the work.

34 The method presented is intended for such cases and it is assumed that leakage conditions will be approximately the same in a commercial installation as those under which the experiment described was made.

DISCUSSION

Wm. G. Christy. In general, this paper appears to be very well written. The authors are to be commended on conducting some experiments and research along this line, as authentic data for use in determining the proper size of dry-vacuum pump to employ seems to be lacking. In commenting on this paper, I wish to call attention to a number of points which are worth considering.

One of the most important phases of the determination of the size of a dry-vacuum pump is the question of volumetric efficien-

cies. The values given in this paper are rather high compared to those used in usual commercial practice. The authors have quoted volumetric efficiencies as high as 95 per cent, while the average values used and quoted commercially vary from 60 per cent to 80 per cent, according to style and size of pump. These experiments were conducted on only one make of pump. The writer believes that the volumetric efficiencies obtained could not safely be used on other styles and makes of pumps. More extensive experiments should be made to obtain results which can be applied in a general manner.

The question of valve ports has not been entered into in this paper. It would be interesting to discuss the construction and characteristics of valve ports and their effects on the size of pump required. As several varieties of valve ports are in commercial use, some data along this line are desirable for proper deductions as to pump sizes.

Nothing is mentioned in this paper in regard to the piston clearances used in this type of pump. What are the actual clearances and what method is used to minimize them? The amount of clearance materially affects the volumetric efficiency and must be taken into account when different makes of pumps are considered.

The authors of this paper have neglected to make allowances for piston-rod displacement in their calculations. This naturally affects the constants and volumetric efficiencies quoted.

Another matter which has apparently not been taken into consideration is the percentage of moisture in the air. This would also incur slight errors in the authors' calculations. While it would appear that the omission of some of these factors would not affect the results very materially, all of them added together might result in quite a discrepancy in the constants and volumetric efficiencies obtained.

Another interesting point which is not covered by this paper is the design of the inlet and discharge valves. How are the valves in the pump under consideration constructed and what pressure is necessary to raise them from their seats? If they are rotary valves, what is the average velocity of the air through the valve ports? The effect of different styles of valves on the volumetric efficiency would be information of value in the matter under discussion.

There is another phase of this matter which has not been taken into account in the preparation of the paper. It has been pretty definitely established that the peak load of a dry-vacuum pump comes on vacua between 12 and 20 in. It usually occurs on vacua of about 16 to 20 in. The weight of the air handled is one of the determining factors and hence the horsepower required for a given installation is governed by the power required to drive the pump when the vacuum is between 12 and 20 in. This is a very important feature to take into account when figuring the size pump required for any given conditions. This is especially true in the case of a steam pump, as the size of the steam end of the pump would be largely determined by the peak-load power requirement.

It has been found that when the vacuum exceeds 26 in., a two-stage pump will give more efficient operation. At least one manufacturer uses a two-stage pump on vacua of 27 in. and above. When a pump of this type is used, a vacuum of about 15 in. is obtained in one stage and the balance in the second stage. With a two-stage pump, smaller piston clearances can be used and larger volumetric efficiency obtained. The results obtained in the authors' experiments are, of course, applicable only to single-stage vacuum pumps. Where high vacua are required, a comparison of the efficiencies, size, cost and performance of single-stage and two-stage vacuum pumps would be valuable.

A. L. McHugh. As this is a subject upon which very few data have been published, the paper is very timely and we wish to contribute our experiences in selecting vacuum pumps particularly for priming centrifugal pumps.

The curves of Fig. 9 show the theoretical strokes of a pump required to reduce a given volume to any negative pressure up to 28 in. From the series of experiments made at our plant with wet vacuum pumps, we found that doubling the theoretical strokes required would give a very close approximation to the correct number.

These curves have proven to be very convenient and save considerable time in selecting pumps for the above mentioned service.

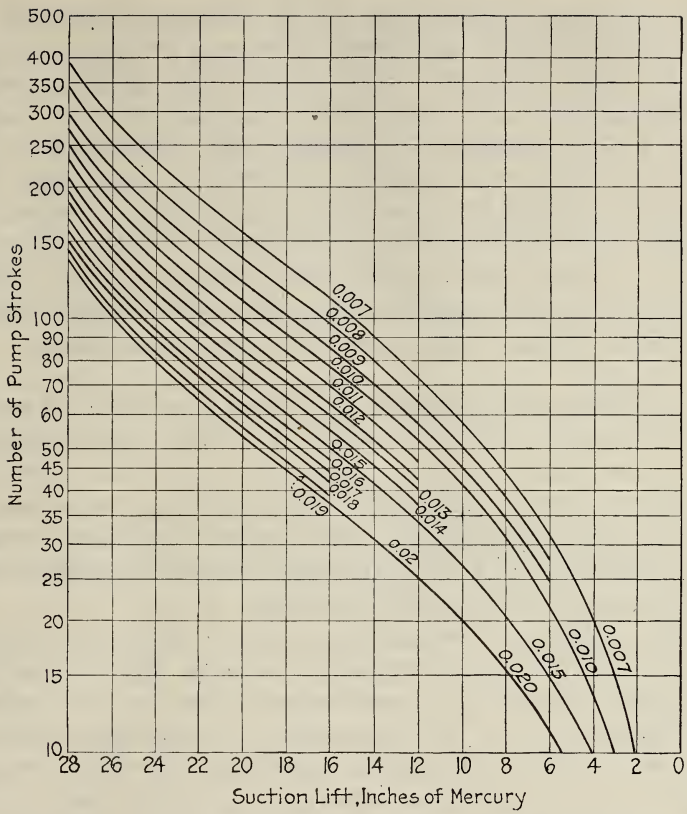

Fig. 9 Theoretical Strokes to Reduce a Given Volume

Curves equal pump displacement per stroke divided by contents of suction piping and pump.

S. A. Moss. The data given in the paper, as far as they go, are a useful contribution to the general subject and a valuable addition to the data already given by Mr. Redfield in the reference mentioned.

There is one point of additional data on which information is needed, which is mentioned in the paper, but which is not complete. This is the matter of leakage. There is an equivalent "leakage area" for every installation. This could be computed by means of suitable formulas from a curve of time against pressure for an evacuated space completely blanked off. If such tests had been made in the authors' work, it seems quite probable that they would have obtained complete correspondence between the average volu-

metric efficiency computations and the time for evacuation. A final formula could be established in terms of average volumetric efficiency and leakage area, and a value from experience inserted for leakage area in any case where it had to be estimated. Leakage area will, of course, vary greatly with tightening up of the joints, or painting on the outside with shellac or asphaltum. Such devices are resorted to in the case of condensers for engines and turbines, and the leakage area is often vastly decreased. The average value in usual cases for remaining leakage is known to manufacturers of dry air pumps for steam plants, and a similar situation undoubtedly exists for the cases which the authors treat.

S. B. Redfield. The writer is pleased to see that the authors have tried out the method of testing dry vacuum pumps which was given in his paper[1] on that subject last year. From a study of this paper it seems to the writer that the lack of agreement between the volumetric efficiency curve applied to the Harris formula, as represented by [2] and [3], must be due to something other than inaccuracy of the nozzle test results.

The problem of determining the size of a vacuum pump to empty a tank to a given vacuum in a given time is one which arises quite frequently and experimental data are very welcome, even though they may be only empiric. In this case, however, it seems to the writer that there is something radically inconsistent because of the fact that the constant K is greater than unity for vacua less than 60 per cent. This constant K is in the nature of a volumetric efficiency, as is shown in Par. 19 of the paper.

In Par. 19 a sample calculation is given to determine the constant and from the Harris formula the theoretical piston displacement should be 0.889 cu. ft. By trial, however, it was found that the pump actually developed 1.17 cu. ft. of piston displacement to get the required vacuum in the given time and the ratio of the theoretical to the actual, or K, is 0.76. It seems to the writer that this is clearly a volumetric efficiency and one would therefore expect it to remain below unity. In Fig. 6 this constant K has been plotted and it is compared in that figure to the average volumetric efficiency obtained by the nozzle test method. It will be noted that the nozzle volumetric efficiency curve quite properly does not exceed 98 per cent, but where the vacuum is less than 60 per cent, the *constant* does become *greater than unity.*

[1] Trans., vol. 43, p. 91.

If it were desired to obtain, say, 30 per cent vacuum in a given size tank in a given time, the process would be to figure the theoretical piston displacement from the Harris formula and then to divide it by the constant, which in this case would be 1.15, because the constant K is the ratio of the theoretical to the actual. This would show that the actual piston displacement would be less than the theoretical and this would make it seem as if possibly the Harris formula is radically at fault. Certainly the volumetric efficiency of the pump can never be greater than unity.

The authors refer to leakage as possibly causing this discrepancy; but certainly leakage into the system would never have the effect of reducing the required size of the pump. Leakage would make it necessary to provide a still larger pump than if there were no leakage.

The Authors. Referring to Mr. Christy's discussion, attention is directed to the fact that the volumetric efficiency of 95 per cent applies only at the beginning of the operation of the pump. This efficiency drops off slowly at first, then rapidly as the vacuum increases, and finally becomes zero when the pump has raised the vacuum to a point where its continued operation fails to increase it further. Note curve in Fig. 2. Volumetric efficiency is affected by two factors, clearance and leakage. Air in the clearance space expands behind the piston until the pressure within the cylinder falls a little below the pressure in the suction pipe before a fresh supply can enter the cylinder. That portion of the cylinder volume traversed by the piston during the interval of re-expansion of the clearance air represents a volumetric loss. However, during the first few revolutions of the pump the air in the suction pipe is at practically atmospheric pressure so there is no appreciable expansion of the air in the clearance space and no reduction in the volumetric efficiency is produced from this source.

Leakage takes place past the piston or through the inlet and discharge valves; but leakage cannot take place without difference of pressure on opposite sides of the piston and valves. There being no pressure differences at the beginning there can be no losses from this source. Theoretically, therefore, the volumetric efficiency of the pump for the first stroke should be 100 per cent. Actually it is not quite that because the pressure in the cylinder has to be raised slightly above atmosphere to overcome the resistance of the

valve springs and the friction of the air in flowing through the discharge valve ports and air passages. Likewise the pressure on the suction side of the piston has to be reduced slightly below the atmospheric pressure in the suction pipe to cause air to flow into the cylinder. The slight reëxpansion and leakage losses created by these small pressure differences prevent the attainment of 100 per cent volumetric efficiency at the instant of starting.

The pumps used in the tests covered by this paper were very similar to the modern single-stage air compressor with horizontal, double-acting cylinders fitted with thin sheet-steel valves in the form of annular rings held against their seats by thin sheet-metal springs. While the unit pressure required to lift the valves from their seats was not measured in the test, it must be very low as the total spring pressure amounts to but a few ounces, being only that necessary to insure prompt seating at the reversal of the motion of the piston. These valves were located in the cylinder heads, thereby avoiding long ports and large clearance volumes. The piston speeds varied from 300 ft. per min. in the smaller sizes to 450 ft. in the larger sizes.

The valve area expressed in terms of piston area was about 4.5 per cent for all sizes tested. The clearance between the piston and the cylinder heads at the end of the stroke was about 0.09 to 0.11 in. in all sizes, or as close as average running conditions would permit with safety.

The volume displaced by the piston rod amounted to only 0.2 of one per cent of the cylinder volume. This error, and that caused by neglect of the moisture content of the air, are so small that their cumulative effect could easily be taken care of by an increase in speed of a revolution or two.

Even if an exact method allowing for every variable to be met in calculating the size of pump to evacuate a definite volume in a given time were available, it would have no great value over one of approximate accuracy. This fact will be apparent when it is considered that after determining the size of pump for a given case by the proposed method, generous allowances must be made for leakages in the vacuum tank or chamber and piping between it and the pump, as the proposed method makes no allowances for leakages outside of the pump cylinder.

The existence of a peak load in the production of a vacuum having a maximum value around 18 in. is known to the authors,

but as the paper contemplated the determination of the pump cylinder size only, no reference to this was made.

With reference to Mr. Moss's discussion, in making the tests described, the space to be evacuated was in the form of a steel tank such as is ordinarily used as a receiver for an air compressor connected to the pump by a short run of pipe. The tank was given a heavy coat of paint while under vacuum and all pipe joints were coated with shellac when they were made up. All errors due to leakage, except those inherent in the pump, were thereby eliminated.

No. 1857

DIESEL-ENGINE CLUTCH USED IN THE GERMAN SUBMARINE "U-117"

By W. H. Nicholson,[1] Camden, N. J.

Non-Member

The purpose of this paper is to give to American builders of Diesel engines the benefit of the author's knowledge of German Diesel-engine clutches obtained at the time of the dismantling of ex-German submarines in the United States. The U-117's clutch is described in detail and other types of German Diesel-engine clutches are covered generally.

THE submarine *U-117* was commissioned by the Germans in 1917 and operated off our North Atlantic coast in that summer and in the summer of 1918. It is known that this submarine took part in the sinking of coal barges and fishing boats in the vicinity of Nantucket and it is supposed that she is responsible for the planting of mines off the Long Island Shore, one of which sank the U. S. Cruiser *San Diego*.[2]

2 The *U-117* is a large mine-laying and -operating submarine. Her overall dimensions are: length, 275 ft.; beam, 17 ft.; and draft, 10 ft. She has a carrying capacity of 20 torpedoes and 45 mines. The armament consists of a 6-in. rapid-fire gun forward and a 5-in. gun aft, both on the main deck. There are four torpedo tubes in the compartment forward, and two mine tubes in the compartment aft.

3 The engine equipment consists of a port and a starboard Diesel oil engine, four-cycle, six-cylinder with two air-compressor cylinders of 1200 i.hp. at 450 r.p.m. for surface cruising and battery charging. The bore of the working cylinder is 17.73 in. and the stroke 16.5 in. The complete engines weigh 57,000 lb. each.

[1] Designer, Engr. Dept., Westinghouse Elec. & Mfg. Co.

[2] The *U-117* was recently used as a target and sunk off the Virginia Capes in the U. S. Army airplane bombing experiments.

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

4 There are also the port and starboard main motors, each rated at 600 hp. at 332 r.p.m., for submerged operation. In general, the design of the motor comprises two compound interpole assemblies in one frame with the armatures in tandem on the same shaft, motor-generator fashion. The motors receive their power from two 124-cell 248-volt storage batteries. When the boat is under

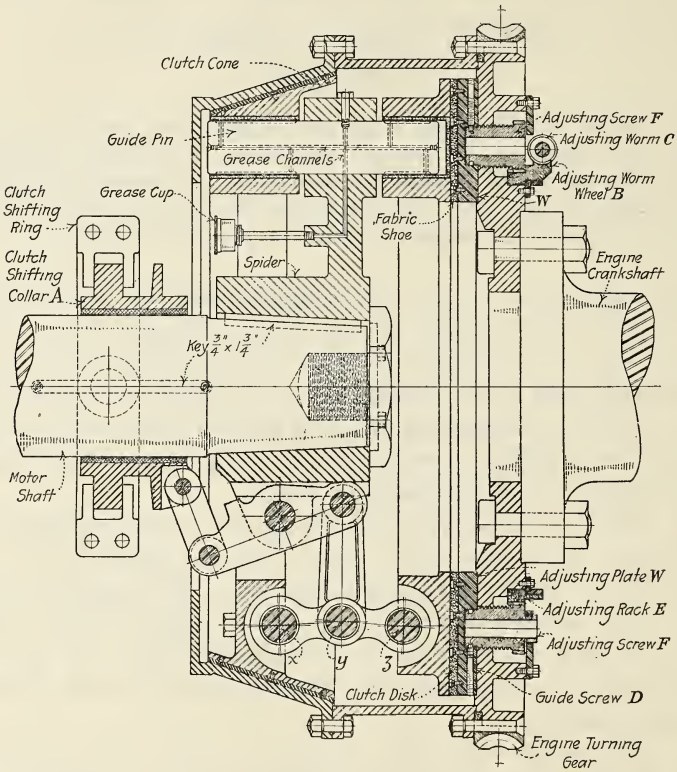

Fig. 1 Sectional View of Diesel-Engine Clutch Used in the German Submarine *U-117*

way on the surface, each motor unit is driven as a generator by the Diesel oil engines, using the excess power of the engine to recharge the batteries and carry the auxiliary power load.

5 The log of the *U-117*, her general description, and the description of her propelling machinery give some idea of the work performed by her clutches.

DETAILS OF THE *U-117* MAIN ENGINE CLUTCH

6 The clutch is located between the Diesel oil engine and the main motor, both port and starboard. It is of the single cone and disk friction dry type, constructed of semi-steel and cast iron, and operated by compressed air. The clutch was built to metric measure. All dimensions given in this paper, however, are in nearest inches or fractions of an inch.

7 The outer casing, adjusting plate, worm wheel, and adjusting screws, which serve as a flywheel, are carried on the engine crankshaft. The casing as shown in Fig. 1 is made of two

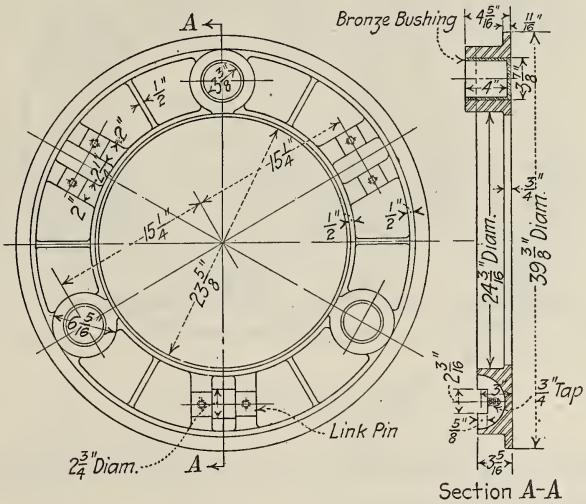

Fig. 2 Clutch Disk

parts bolted together. The inner surface of the after casing is machined to a cone surface; the forward casing is a straight sleeve or spacer for power transmission only. The extreme outside diameter of the casing is 45 in.

8 The inner clutch cone or male cone and clutch disk are carried on the guide pins of the spider, on which they are a sliding fit. They are of semi-steel and the outer surface of the male cone is finished to conform to the conical surface of the female or flywheel part. The forward surface of the clutch disk is finished for contact on the fabric shoe of the adjusting plate. Figs. 2 and 3

are detail sketches of the male cone and the clutch disk carried by the spider. The taper of the cone surfaces is about 14 deg. The cone and disk are bronze-bushed for the guide pins.

9 To the outer surface or periphery of the male cone there is attached, by means of countersunk machine screws, a fiber shoe as shown in Figs. 1 and 3. The shoe is $\frac{1}{4}$ in. thick and is made up in four sections, the divisions being in the direction of the axis of the guide pins or shaft. It is the surface of the fiber shoe

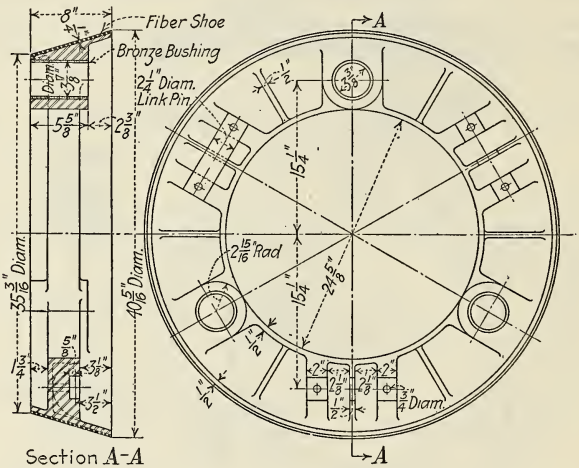

Fig. 3 Clutch Cone

that contacts with the metal surface of the female or outer casing when the clutch is thrown in.

10 The spider, attached to the motor shaft by a single key and a retaining screw has three guide pins parallel to the axis of the engine and motor shafts and extending through the spider on either side as shown in Figs. 1 and 4. The guide pins are steel forgings and are shrunk in the spider.

11 The clutch is thrown in by forcing the male cone and the disk apart until the male and female cone surfaces engage and the disk surface engages the fabric shoe of the adjusting plate; and by drawing the male cone and the disk together and thus releasing the surfaces the clutch is thrown out. The male cone and the disk are drawn together or forced apart by a series of links and toggles actuated by a sleeve or collar (*A* in Fig. 1) sliding on the

motor shaft. The shifting collar is shown in detail in Fig. 5. It is babbitted for a sliding fit on the motor shaft and is single-keyed thereto.

12 Referring to the lower half of Fig. 1, the center x is shown fixed to the male cone and center z to the disk. The clutch is en-

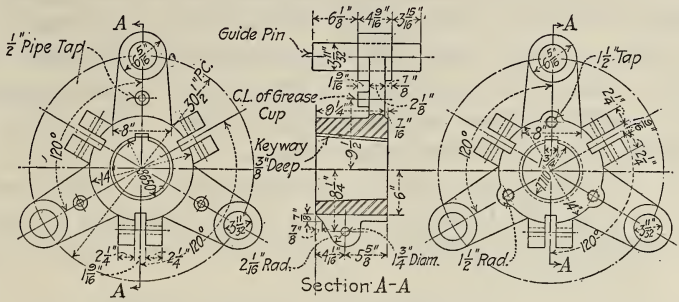

Fig. 4 Clutch Spider

gaged by separating centers x and z and by forcing center y up to a line adjoining centers x and z. There are three sets of operating links and toggles equally spaced between guide pins of the spider as shown in Fig. 4.

13 To the after side of the worm wheel there is attached by means of six adjusting screws a steel disk or adjusting plate faced

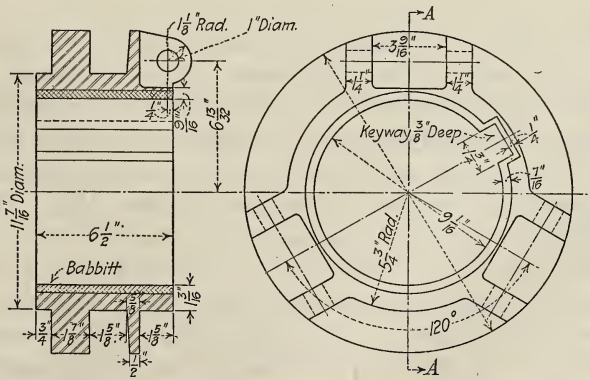

Fig. 5 Clutch-Shifting Collar

with fabric. The fabric is $\frac{7}{16}$ in. thick and serves the same purpose as the fiber shoe of the male cone. The fabric is also in four sections and is secured by means of countersunk machine screws.

The sectional divisions of the fabric are radial. When the clutch is engaged, the forward surface or finished surface of the clutch disk has contact with the fabric shoe.

14 The fore-and-aft movement of the male cone and disk is $\frac{3}{8}$ in., or a total of $\frac{3}{4}$ in. The fiber and fabric shoes have some flexibility and it is these surfaces that absorb the load shocks and transmit the power of the engine. The design of this clutch differs in this respect from the clutch used with the 3000- and 1750-hp. engines, in that those clutches have, when engaged, a metal-to-metal contact on lubricated surfaces. The fabric and fiber shoes may be renewed when worn.

15 The adjusting worm wheel is carried on the engine turning gear (see *B* in Fig. 1) and is rotated by the adjusting worm (*C*) which is hand-operated from either end of the worm shaft by means

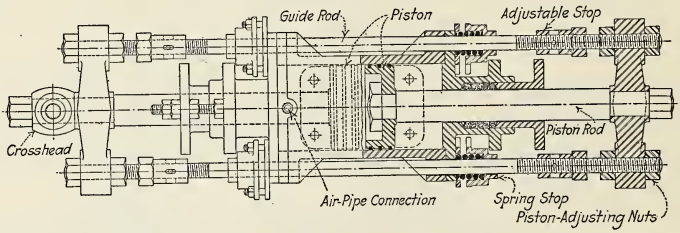

Fig. 6 Clutch-Shifting Cylinder

of a wrench. The adjusting worm is bracketed to the engine turning-gear cover plate. Any movement of the adjusting worm is transmitted through the rack (*E*) and rotates the adjusting screws (*F*) which are threaded through the worm wheel. There are six adjusting screws equally spaced, three of which are secured to the adjusting plate by means of the guide screws (*D*). The other three adjusting screws have contact only on the forward surface of the adjusting plate; their ends are flat and do not penetrate the plate. The sketches do not show the short adjusting screws. By means of the adjusting mechanism there is a movement of $\frac{9}{16}$ in. between the disk and the fabric shoe, the adjusting plate moving aft from the turning gear that distance at the surfaces marked *W* in Fig. 1.

16 The guide pins of the spider are lubricated by means of three grease cups attached to its arms by a length of pipe. The guide pins and spider arms are drilled for the passage of grease

and the former are grooved for its distribution. Plugs are inserted in the blind ends of drilled holes. The circumferential force or centrifugal force of the clutch should carry sufficient grease to the pins without attention to the grease cups except for occasional filling.

17 The clutch shifting cylinder shown in Fig. 6 is of semi-steel. The bore is 7 in. and the stroke is from 7½ to 8 in., depending on the setting of the pistons. The pistons may be set for long or short stroke by means of the piston adjusting nuts. To prevent pistons from striking cylinder heads when operating, adjustable stops are provided on the guide rods and spring stops are

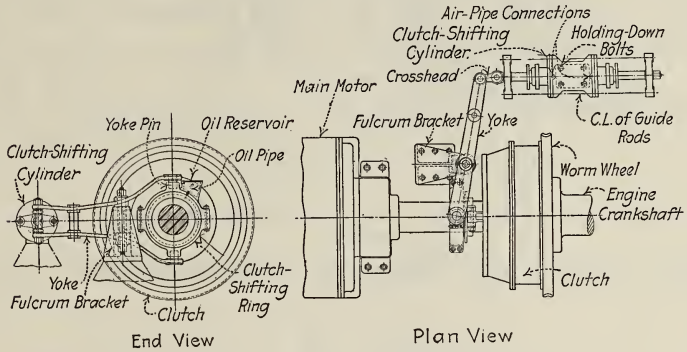

Fig. 7 General Arrangement of Clutch

secured to the cylinder heads. The springs at either end of the cylinder absorb the shocks when throwing the clutch in or out.

18 The general arrangement of the clutch is shown in Fig. 7. The clutch operating cylinder is placed alongside the engine with its center line parallel with the axis of the crankshaft.

19 The clutch cylinder is operated by air pressure and transmits its power through the crosshead, yoke, and fulcrum to the clutch.

20 The fulcrum bracket is of semi-steel and is carried on a foundation from the ship's floor. It is estimated that the load at the fulcrum is about 16,000 lb.

21 The clutch shifting ring is held in its position by the yoke pins and is lubricated for contact with the clutch shifting collar by means of a wick in the oil reservoir.

22 The engine-turning gear carried with the clutch consists of a worm wheel carried on the engine crankshaft flange and a

hand-operated worm which is locked out of position when not in use. The diameter of the turning gear is 46⅜ in., which is the extreme outside dimension of the assembled clutch.

23 Fig. 8 shows diagrammatically the arrangement of the clutch-shifting piping. Compressed air for the operation of the clutch is supplied from the main engine air-starting flasks at 160 atmospheres (metric measure) or about 2275 lb. per sq. in. It is noted that the air is drawn from the top of the flasks by means of an internal pipe to prevent water that may be collected in the

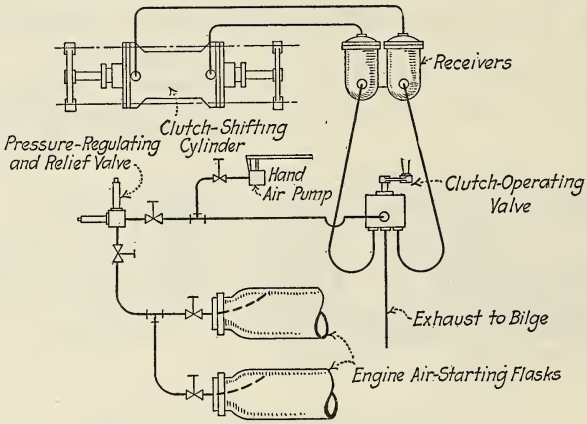

Fig. 8 Diagram Showing Arrangement of Clutch-Shifting Piping

flasks from passing through to the shifting cylinder and damaging it.

24 There is a pressure-regulating and relief valve in the pipe from the air-starting flasks, with stop valves on either side and at the flasks. The air passes through the clutch-operating valve to the receivers and then to the clutch-operating cylinder.

25 The clutch-operating valve is so designed that its ports take care of the exhaust air from the opposite end of the clutch-operating cylinder by a single movement of the handle, exhausting the air and entrained water to the ship's bilges.

26 When the clutch is thrown in, it is self-locking, as shown by position of links and toggles in lower half of Fig. 1. It is to be noted that this locking is accomplished by forcing center y slightly above centers x and z. Air may then be shut off from the

clutch-shifting cylinder until it is necessary to throw the clutch out. The centrifugal force of the clutch will not throw it out. The advantage of the self-locking feature lies in the assurance that the holding power of the clutch does not depend on a constant pressure of air in the cylinder. Where air pressure would be used for holding the clutch in, there would be the possibility of leaks or fluctuations and the release of the clutch.

27 All air piping is ½ in. in inside diameter and of seamless drawn steel with steel rings brazed to the ends for making up the joints. The pipes are secured in forged-steel fittings by means of a male nut placed on the pipe back of the brazed ring, the nuts then being screwed into the female joint of the fittings. Gaskets are either fiber or copper, placed within the fitting. All pipes and fittings tested to 3000 lb. hydrostatic pressure per sq. in.

28 For emergency there is a hand pump for the operation of the clutch.

CLASSIFICATION OF GERMAN DIESEL ENGINES AND CLUTCHES

29 The German Diesel oil engines were generally classified as follows:

10-cylinder, 4-cycle, 3000 i.hp., for large cruisers (submarine)
6-cylinder, 4-cycle, 1750 i.hp., for large operating submarines
6-cylinder, 4-cycle, 1200 i.hp., for large mine-laying and operating submarines
6-cylinder, 4-cycle, 550 i.hp., for mine-laying and coastal submarines.

30 Clutches for the 3000- and 1750-i.hp. engines were of the double-cone, friction, lubricated type, operated by compressed air or electric motor.

31 Clutches for the 1200- and 500-i.hp. engines were of the single-cone and disk, friction, dry type. The 1200-i.hp. engine clutches were operated by air and are described in detail in this paper. The 550-i.hp. engine clutches were operated by hand through worm and gear.

32 The details of design of the German Diesel-engine clutches were practically unknown in this country prior to the surrender of the German submarines, and considerable difficulty was experienced in dismantling them.

33 At the time of the arrival of the submarine freighter *Deutschland* at New London, when an appointed board of naval officers and civilians inspected her to establish her status, it is

known that much interest was shown in her clutches, the construction of which was considered remarkable in so far as so small a diameter of clutch was capable of transmitting such large horsepower. No information of value was then obtainable, but later, when the surrendered German submarines arrived in this country, the clutches were one of the first mechanisms to be investigated.

34 It is believed that many improvements can be made in the design of American clutches by adopting some of the principles of the German clutches. The author has had some experience with American clutches and has found them to be of the tooth or positive-drive type, or of the friction-jack type. These clutches have been the source of trouble and are constantly in need of repair. The tooth type does not allow the flexibility of shaft speeds required in maneuvering a ship that a friction type allows, and from experience it has been found that the friction types are not of sufficient strength and rigidity to transmit heavy loads. During the war one of our American submarines went below her depth by accident and it was necessary to insert emery paper in the clutch before the jaws would take hold and allow her motors to operate the shaft-driven bilge pump. Such delays may cause the loss of a ship and many lives.

35 The friction-jack types are usually thrown in by hand through a series of levers and therefore lack the holding power of an air-operated clutch.

36 With the friction-jack type of clutch there is nearly always a misalignment of shafts when the clutch is in, due to improper setting of the jack screws or adjusting screws. The screws are placed radially and must be adjusted from time to time to take up wear. It requires a skilled operator to properly adjust such a clutch.

37 In the German clutches the adjustment is fore and aft along the axis of the shaft, and with a proper alignment of shafts when machinery is first installed there is little possibility of misalignment when the clutch is in.

38 The author will be glad to give any further information desired by clutch manufactures or engineers that will help to improve our American clutches.

No. 1858

STRESSES IN CYLINDRICALLY SHAPED ROTORS OF UNIFORM DIAMETER

By C. M. Laffoon,[1] East Pittsburgh, Pa.
Non-Member

The author develops a general theory and analyzes mathematically the stresses in cylindrical rotors of uniform diameter such as are commonly associated with different classes of electrical machinery. The derivation of the fundamental equations is based upon the method used by Stodola in analyzing stresses in rotating disks of steam turbines. The general equations are derived and applications to seven different types of cylindrical rotors are analyzed and the formulas for these cases derived.

A KNOWLEDGE of the magnitude of the rotational stresses due to centrifugal force is of prime importance to engineers who are concerned with either the design or operation of high-speed rotating machinery, such as steam turbines, turbo-alternators, high-frequency generators, gyroscopes, and various other classes of electrical machines. The problem of determining the magnitude of the stresses is of especial interest to the designing engineer who is responsible for fixing the physical dimensions and proportions of particular machines which operate at given speeds.

2 The general theory and mathematical analysis of the stresses in rotating disks, as applied to steam turbines, is admirably treated in Stodola's classical treatise, The Steam Turbine. This analysis, however, to the author's knowledge, has never been extended to cover the types of rotors which are commonly associated with the different classes of electrical machinery. In fact, mechanical handbooks are practically the only available source of information on this subject and the data contained therein are usually very brief and applicable only to special cases. There has been a long-felt need by

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electrical engineers for a more complete and comprehensive analysis of the stresses in the rotating elements of electrical machines than is usually given in handbooks. In view of this fact it is the purpose of the author in the present discussion to present the physical conception and analysis of the stresses in several of the most important cases of cylindrically shaped rotors of uniform diameter. The derivation of the fundamental equations is based on the method used by Stodola and is given only as a matter of reference so that the reader can follow the discussion without referring to any other work. In order to make the physical conception of the analysis of the centrifugal stress phenomenon as clear as possible, two special elementary cases will be discussed before considering the actual cases of cylindrical rotors.

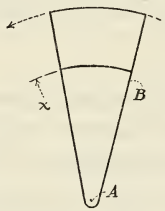

Fig. 1

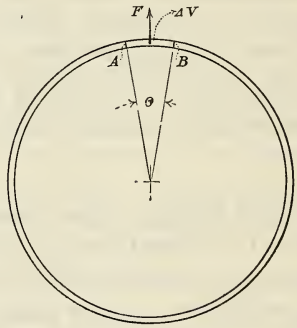

Fig. 2

ANALYSIS OF THE STRESSES

3 If the rotating body consists of an isolated section B of a solid cylindrical rotor, Fig. 1, which rotates about an axis A at an angular velocity w, the total centrifugal force of the body acts in a radial direction and must be supported by the axis at A. At any distance x from the axis the centrifugal force of the external portion must be carried by the section of the material at this point. Hence the centrifugal force produces a radial stress in the material which varies in value from zero at the external radius to a maximum at the axis. The radial stress in the material causes it to elongate or stretch in the radial direction. Since the elongation is proportional to the stress, the radial elongation per unit length will be a maximum at the axis and at other points it will be proportional to the corresponding stress at these points.

4 On the other hand, if the isolated section consists of a thin annular cylinder as shown in Fig. 2, the centrifugal force of any incremental volume ΔV acts radially but has nothing to support it in this direction. Therefore it must be supported by tangential forces in the material adjacent to the radial faces A and B. In order for the elementary volume to be in equilibrium, the radial components of the tangential forces which act at right angles to the radial faces A and B must balance the centrifugal force F, as shown in Fig. 3. The tangential stress in the material produces an elongation in the circumference and a reduction in the radial thickness of the cylindrical shell. Hence in this case the centrifugal force is counterbalanced by tangential stresses which tend to increase the diameter of the annular cylinder. The percentage increase is greater for the internal diameter than for the external diameter on account

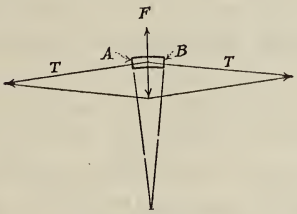

Fig. 3

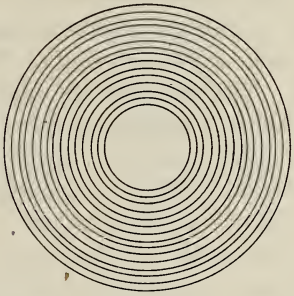

Fig. 4

of the decrease in thickness of the cylinder walls. It is obvious, then, that if the rotor were composed of annular laminæ, as shown in Fig. 4, each ring would act independently in counterbalancing the centrifugal force acting on it. Since the centrifugal force and the mass of the laminæ vary directly as the radius, the tangential stresses in the laminæ increase approximately as the square of the radius. Similarly, the radial expansion is a maximum for the outside laminæ and decreases approximately as the square of the radius for the internal laminæ.

5 The solid or hollow cylindrical rotor of appreciable radial thickness is in reality a combination of the two cases previously considered. The centrifugal forces of the circumferential sections are the same as for the laminated rotor, but the sections are not free to expand independently and thereby counterbalance their own respective centrifugal forces. Since the centrifugal force of the cir-

cumferential section varies as the square of the radius and all of the sections are elastically coupled together, part of the centrifugal force of the outermost section will be transmitted to the adjacent inner section as an external radial load. This additional external load on the inner section produces radial and tangential stresses in it which add to the corresponding stresses due to its own centrifugal force. This section, in turn, transmits a greater external radial load to the next adjacent inner section than it would if the outermost section were not present. This loading-back effect cumulatively increases in passing from the external to the internal sections, and consequently the radial and tangential stresses are higher for the internal sections than for the external sections. It is obvious in the case of a solid rotor that the radial and tangential stresses increase from zero and minimum values, respectively, at the external radius to maximum and equal values at the center. Similarly, in the case of the hollow cylindrical rotor the tangential stress increases from a minimum value at the external radius to a maximum value at the internal radius. The radial stress increases from zero at the external radius and reaches a maximum value, and then decreases and reaches zero at the internal radius. In both cases the greatest stress occurs at the internal radius and consequently this is the point at which rupture or failure usually occurs. It is also obvious from the above analysis that with a central hole in the rotor the internal radial stress is zero and its equivalent effect must be compensated for by an increased tangential stress at this point. Since the two stresses are normally equal at the center and the radius and tangent of a center point are identical, a pinhole at the center would reduce the radial stress to zero and consequently double the tangential stress.

6 It is obvious from the above consideration, neglecting the stresses due to shear and bending, that any incremental volume, ΔV, of the rotor, Fig. 5, is subjected to the following forces due to rotation:

a The centrifugal force of the elementary volume, which acts outward in the radial direction

b The tangential forces which act at right angles to the radial faces A and B, and have radial components directed inward

c Radial forces which tend to separate the elementary volume from the main body of the rotor at the inner and

outer circumferential surfaces of the elementary volume. The difference between these two forces determines the magnitude of the force acting on the elementary volume; and the relative magnitude of the forces determines the direction of the resultant. If the radial force at the outer surface is the greater, the direction of the resultant will be outward. Similarly, if the radial force at the inner surface is the greater, the resultant force acting on the elementary section will be directed inward. In order for the incremental volume to be in stable equilibrium, the algebraic sum of the radial components must be zero.

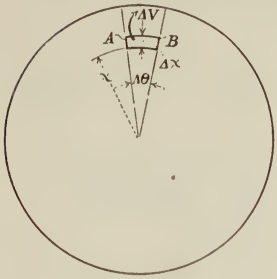

Fig. 5

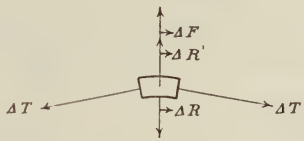

Fig. 6

DERIVATION OF THE GENERAL EQUATIONS

7 In Figs. 5 and 6 let —

x = radial distance of the elementary section from the rotor axis

l = length of rotor (constant)

S_t = tangential stress acting at right angles to the radial faces of the elementary section

S_r = radial stress at the inner circumference of the section

W = weight per unit volume of the rotor material

g = acceleration due to gravity in feet per second per second

w = angular velocity of the rotor in radians per second

v = contraction coefficient (Poisson's ratio).

Then from Figs. 5 and 6,

$\Delta V = x\Delta\theta l \cdot \Delta x$ = volume of the elementary section

$\Delta M = \dfrac{W}{g} lx \cdot \Delta\theta \cdot \Delta x$ = mass of the elementary section

$\Delta F = \Delta M \times \frac{w^2 x}{12}$ = centrifugal force acting on the elementary section

$= \frac{W}{g} lw^2 \frac{x^2}{12} \Delta\theta \cdot \Delta x$, where x is in inches

$\Delta T = S_t l \Delta x$ = tangential force acting across the radial faces or A and B

$\Delta R = S_r l x \cdot \Delta\theta$ = radial force at the inner circumference of the section

$\Delta R' = (S_r + \Delta S_r) \cdot (x + \Delta x) \cdot l \cdot \Delta\theta$ = radial force at the outer circumference of the section

ΔZ = radial component of the tangential forces

$= 2\Delta T \sin \frac{\Delta\theta}{2}$

$= \Delta T \cdot \Delta\theta$ (approximately)

$= S_t\, l \cdot \Delta x \cdot \Delta\theta$

In order for the elementary section to be in equilibrium —

$$\Delta F + \Delta R' - \Delta R - \Delta Z = 0$$

or $$\frac{Ww^2 lx^2}{12g} \cdot \Delta\theta \cdot \Delta x + (S_r + \Delta S_r)\,(x + \Delta x)\; l \cdot \Delta\theta - S_r l\, x \cdot \Delta\theta - S_t l \cdot \Delta\theta \cdot \Delta x = 0$$

Simplifying this equation and neglecting the differentials of the second order, gives —

$$\frac{ds_r}{dx} + \frac{l}{x}(S_r - S_t) + \frac{W}{12g} w^2 x = 0 \quad . \; . \; . \; . \; . \; . \quad [1]$$

In order to solve this differential equation, it is convenient to express S_r and S_t in terms of the radial elongation.

Let λ = total radial elongation of a point originally at a distance, x, from the axis, Figs. 7(b) and 7(c)

$\lambda + \Delta\lambda$ = total radial elongation of a point originally at a distance, $x + \Delta x$, from the axis, Figs. 7(b) and 7(c)

E_r' = radial expansion per unit length of the radius

E_t' = tangential expansion per unit length of the circumference.

The radial expansion due to S_r is $\frac{S_r}{E}$, where E is the modulus of elasticity of the rotor material. The tangential stress acting simul-

taneously produces a contraction $\frac{vS_t}{E}$. The resultant radial and tangential expansions per unit length due to the stresses S_r and S_t are —

$$\left.\begin{aligned} E_r' &= \frac{1}{E}(S_r - vS_t) \\ E_t' &= \frac{1}{E}(S_t - vS_r) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [2]$$

The values of E_r and E_t can also be found in terms of the rotor dimensions by considering the length of the radius and circumference before and after expansion, as shown in Figs. 7(*a*), 7(*b*), and 7(*c*).

$$\frac{E_t' = 2\pi(x + \lambda) - 2\pi x}{2\pi x} = \frac{\lambda}{x} \quad \ldots\ldots\ldots [3]$$

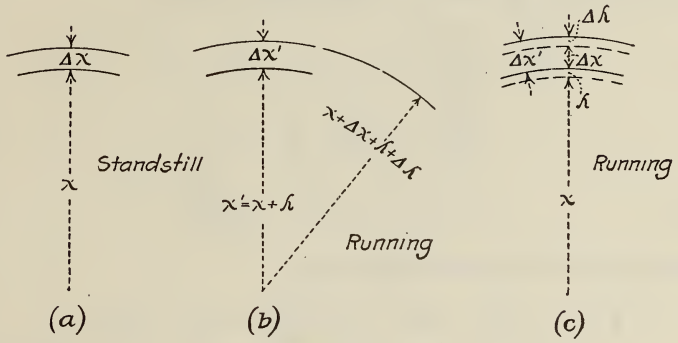

FIG. 7

8 From Fig. 7(*c*) the radial elongation of the length Δx is $\Delta\lambda$, hence the elongation per unit length —

$$E_r' = \frac{\Delta\lambda}{\Delta x} = \frac{d\lambda}{dx} \quad \ldots\ldots\ldots\ldots [4]$$

Substituting these values of E_r' and E_t' in Equations [2] and solving for S_r and S_t gives —

$$\left.\begin{aligned} S_r &= \frac{E}{1-v^2}\left(\frac{v\lambda}{x} + \frac{d\lambda}{dx}\right) \\ S_t &= \frac{E}{1-v^2}\left(\frac{\lambda}{x} + v\frac{d\lambda}{dx}\right) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [5]$$

Taking the derivative of S_r with respect to x gives —

$$\frac{ds_r}{dx} = \frac{E}{1-v^2}\left(\frac{vx\frac{d\lambda}{dx} - \lambda v}{x^2} + \frac{d^2\lambda}{dx^2}\right) \quad \ldots\ldots\ldots [6]$$

Substituting $\frac{ds_r}{dx}$, S_r, and S_t in Equation [1] and simplifying gives —

$$\frac{d^2\lambda}{dx^2} + \frac{1}{x}\frac{d\lambda}{dx} - \frac{\lambda}{x^2} + Ax = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where $A = \frac{W}{12g} w^2 \frac{(1 - v^2)}{E}$

The term involving x can be eliminated by substituting —

$$\lambda = Z + ax^3$$

$$\frac{d\lambda}{dx} = \frac{dZ}{dx} + 3\,ax^2, \text{ and } \frac{d^2\lambda}{dx^2} = \frac{d^2Z}{dx^2} + 6\,ax$$

Substituting $\frac{d^2\lambda}{dx^2}$, $\frac{d\lambda}{dx}$ and λ in Equation [7] gives —

$$\frac{d^2Z}{dx^2} + \frac{1}{x}\frac{dZ}{dx} - \frac{Z}{x^2} + (8a + A)x = 0$$

or

$$\frac{d^2Z}{dx^2} + \frac{1}{x}\frac{dZ}{dx} - \frac{Z}{x^2} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

where

$$a = -\frac{A}{8} = \frac{-W}{12g} w^2 \frac{(1 - v^2)}{8\,E}$$

To solve this differential equation, let —

$$Z = bx^n, \frac{dZ}{dx} = n\,bx^{n-1}, \text{ and } \frac{d^2Z}{dx^2} = n(n-1)\,bx^{n-2}$$

Substituting $\frac{d^2Z}{dx^2}$ in Equation [8] and simplifying, gives —

$$n^2 = 1$$

therefore —

$$n = \pm 1$$

and the general solution is —

$$Z = b_1x + b_2x^{-1}$$

Hence the general expression for the radial elongation is —

$$\lambda = ax^3 + b_1x + b_2x^{-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

$$\frac{\lambda}{x} = ax^2 + b_1 + b_2x^{-2}$$

$$\frac{d\lambda}{dx} = 3ax^2 + b_1 - b_2x^{-2}$$

Substituting $\frac{\lambda}{x}$ and $\frac{d\lambda}{dx}$ in Equations [5] and simplifying,

$$\left.\begin{aligned} S_r &= \frac{E}{1-v^2}\left(ax^2(3+v) + b_1(v+1) + b_2(v-1)\,x^{-2}\right) \\ S_t &= \frac{E}{1-v^2}\left(ax^2(3v+1) + b_1(v+1) - b_2(v-1)\,x^{-2}\right) \end{aligned}\right\} \quad [10]$$

9 The constants of integration b_1 and b_2 can be determined from the initial conditions for rotors of different specifications.

APPLICATION TO DIFFERENT TYPES OF CYLINDRICAL ROTORS

10 *Case I. Solid Cylindrical Rotor Without Central Hole*— The general equations for the radial and tangential stresses and the radial elongation in a cylindrical rotor due to the action of the

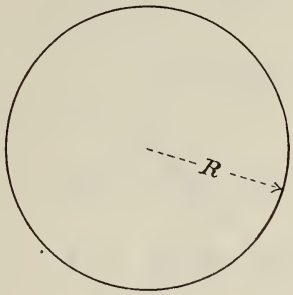

FIG. 8

centrifugal force are, from Equations [9] and [10],

$$S_r = \frac{E}{1-v^2}\left[ax^2(3+v) + b_1(1+v) + b_2(v-1)x^{-2}\right]$$

$$S_t = \frac{E}{1-v^2}\left[ax^2(3v+1) + b_1(1+v) - b_2(v-1)x^{-2}\right]$$

$$\lambda = ax^3 + b_1x + b_2x^{-1},$$

where S_r = radial stress in lb. per sq. in. at radius x

S_t = tangential stress in lb. per sq. in. at radius x

λ = radial elongation in inches at radius x

$E = 29 \times 10^6$ = Young's modulus of elasticity

$v = 0.3$ = Poisson's ratio, or contraction coefficient for steel

b_1, b_2 = constants of integration

$$a = -\frac{W}{12g}\,w^2\,\frac{(1-v^2)}{8\,E}$$

$W = 0.285$ lb. per cu. in. for the rotor material

$g = 32.2$ = acceleration due to gravity in ft. per sec. per sec.

w = angular velocity in radians per sec.
x = radius in inches.

11 The constants of integration b_1 and b_2 can be determined from the limiting conditions: namely,

$S_r = 0$, when $x = R$, where R is the external radius, and
$\lambda = 0$, when $x = 0$.

Therefore b_2 must be zero if these conditions be true. Applying these conditions to the above equations gives —

$$b_1(1 + v) = -aR^2(3 + v) \quad \text{[10b]}$$

$$\lambda = ax^3 + b_1x \quad \text{[10c]}$$

From Equation [10b] —

$$b_1 = -aR^2 \frac{3 + v}{1 + v}$$

therefore —

$$\lambda = a\left[x^3 - R^2x \frac{3 + v}{1 + v}\right]$$

But

$$a = \frac{-W}{12g} w^2 \frac{1 - v^2}{8E}$$

or

$$\lambda = \frac{W}{12g} w^2 \frac{1 - v^2}{8E} \left[R^2x \frac{(3 + v)}{1 + v} - x^3\right]$$

but $w = \dfrac{V_x}{x}$ where V_x = peripheral speed at radius x in inches per sec.

Then —

$$\lambda = \frac{W}{12g} \frac{V_x^2}{x^2} \frac{1 - v^2}{8E} \left[R^2x \frac{3 + v}{1 + v} - x^3\right]$$

$$= \frac{W}{12g} V_x^2 \frac{1 - v^2}{8E} \left[\frac{R^2}{x} \frac{3 + v}{1 + v} - x\right]$$

For steel —

$$\frac{W}{12g} = \frac{490}{1728} \times \frac{1}{12} \times \frac{1}{32.2}$$

$$E = 29 \times 10^6$$

$$v = 0.3$$

$V_x = 12\, V_{fx}$, where V_{fx} is peripheral speed in ft. per sec. at radius x.

For $x = R$,

$$\lambda = \frac{490}{1728} \times \frac{1}{12} \times \frac{1}{32.2} \frac{(1 - 0.3^2)}{8 \times 29 \times 10^6} \times 12^2 \left[\frac{3 + 0.3}{1 + 0.3} - 1\right] RV_{fR}^2$$

$$\lambda = 0.64 \times 10^{-9}\, R V_{fr}^2,$$

where R = external radius in inches and V_{fr} = peripheral speed in ft. per sec. at R. For any value of x, the radial elongation is —

$$\lambda = 0.64 \times 10^{-9}\, V_{fr}^2\, x \left(1.65 - 0.65 \frac{x^2}{R^2}\right)$$

12 Substituting these values of b_1, b_2 and a in Equations [1] and [2],

$$\left.\begin{array}{l} S_t = 0.043606\, V_{fr}^2 \left(1 - 0.57575 \dfrac{x^2}{R^2}\right) \text{ at any radius } x \\ S_t = 0.0185\, V_{fr}^2 \text{ at the external radius, i.e., } x = R \\ S_t = 0.043606\, V_{fr}^2 \text{ at center of disk, i.e., } x = 0 \end{array}\right\} \quad \ldots [11]$$

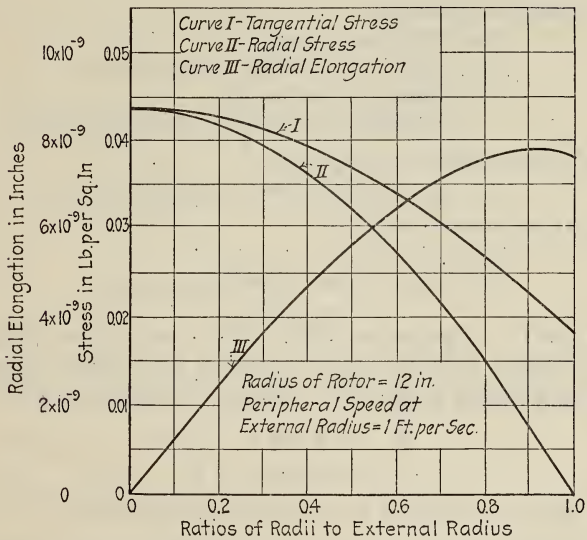

FIG. 9 STRESSES AND RADIAL ELONGATION OF SOLID CYLINDRICAL ROTOR

$$\left.\begin{array}{l} S_r = 0.043606\, V_{fr}^2 \left(1 - \dfrac{x^2}{R^2}\right) \text{ at any radius } x \\ S_r = 0 \text{ at the external radius, i.e. } x = R \\ S_r = 0.043606\, V_{fr}^2 \text{ at center of disk, i.e., } x = 0 \end{array}\right\} \quad \ldots\ldots\ldots [12]$$

where V_{fr} is the peripheral speed in ft. per sec. at the external radius, and both x and R are expressed in inches.

13 Curves I and II of Fig. 9 show the values of the tangential and radial stresses in a solid cylindrical rotor for a peripheral speed,

at the external radius, of 1 ft. per sec. The stress is expressed in lb. per sq. in. and the radii are expressed as decimal fractions of the external radius. Both stresses are a maximum and equal at the axis of the rotor. Curve III shows the radial elongation in inches for a solid cylindrical rotor which has a radius of 12 in. and an external peripheral speed of 1 ft. per sec. The elongation reaches a maximum value at a radius equal to approximately 92 per cent of the external radius. The decrease in the radial elongation for radii greater than 92 per cent of the external radius is due to the fact that the radial contraction of the material at these radii caused by the tangential stress is greater than the radial expansion produced by the radial stress.

14 Numerical Example:

$$R = 20 \text{ in. external radius}$$
$$V_{fr} = 400 \text{ ft. per sec. at the external radius}$$
$$\lambda = 0.64 \times 10^{-9} \times 20 \times 400^2$$
$$= 0.00205 \text{ in. elongation at the external radius.}$$

The stresses at the center of the rotor are

$$S_t = S_r = 0.043606 \times 400^2 = 6975 \text{ lb. per sq. in.}$$

The stresses at the external radius are —

$$S_r = 0$$
$$S_t = 0.0185 \times 400_2 = 2960 \text{ lb. per sq. in.}$$

15 *Case II. Cylindrical Rotor with Central Hole* (Fig. 10). — This case applies particularly to rotating cylindrical drums, plate flywheels, and pulley rims. The limiting conditions are —

$$S_r = 0 \text{ when } x = r, \text{ and}$$
$$S_r = 0 \text{ when } x = R$$

Applying these conditions to Equation [**10**] gives —

$$0 = ar^2 (3 + v) + b_1 (v + 1) + \frac{b_2}{r^2} (v - 1)$$

$$0 = aR^2 (3 + v) + b_1 (v + 1) + \frac{b_2}{R^2}(v - 1)$$

Solving these simultaneous equations for b_1 and b_2 gives —

$$b_1 = \frac{-a(3 + v)(r^2 + R^2)}{v + 1}$$

$$b_2 = \frac{a(3 + v) r^2 R^2}{v - 1}$$

16 The values of the stresses and radial elongation can be found by substituting the values of b_1 and b_2 in Equations [**9**] and [**10**].

$$S_t = 0.043606\ V_{fr}^2 \left[\frac{r^2}{R^2} + 1 + \frac{r^2}{x^2} - \frac{x^2}{R^2} 0.57575\right] \text{at any radius } x \quad [13]$$

$$S_t = 0.043606\ V_{fr}^2 \left[0.42424\ \frac{r^2}{R^2} + 2\right] \text{at } x = r \quad [14]$$

$$S_t = 0.043606\ V_{fr}^2 \left[\frac{2r^2}{R^2} + 0.42424\right] \text{at } x = R \quad [15]$$

$$S_r = 0.043606\ V_{fr}^2 \left[\frac{r^2}{R^2} + 1 - \frac{x^2}{R^2} - \frac{r^2}{x^2}\right] \text{at any radius } x \quad [16]$$

$$S_r = 0 \text{ at } x = r \text{ and } x = R \quad [17]$$

$$\lambda_R = 0.64 \times 10^{-9}\ V_{fr}^2\ R \left[1 + 4.71\ \frac{r^2}{R^2}\right] \text{at } x = R \quad [18]$$

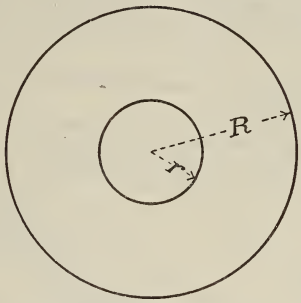

Fig. 10

$$\lambda_x = 0.64 \times 10^{-9}\ V_{fr}^2\ x \left[1.65 + 1.65\ \frac{r^2}{R^2} - 0.65\ \frac{x^2}{R^2} + 3.06\ \frac{r^2}{x^2}\right]$$
$$\text{at } x = x \quad [19]$$

$$\lambda_r = 0.64 \times 10^{-9}\ V_{fr}^2\ r \left[4.71 + \frac{r^2}{R^2}\right] \text{at } x = r \quad [19a]$$

17 The curves in Fig. 12 show the variation of the radial and tangential stresses for hollow cylindrical rotors with different ratios of internal to external diameters. The stresses are based on a peripheral speed at the external surface of 1 ft. per sec. The radial elongation is based on a rotor with a 12-in. radius and a peripheral speed at the external radius of 1 ft. per sec.

18 Numerical Example:

R = 20 in., external diameter of rotor
r = 12 in., internal diameter of rotor
$\frac{r}{R}$ = 0.6, ratio of internal to external diameter
V_{fr} = 400 ft. per sec., peripheral speed at external radius.

From Figs. 11 and 12 the stresses for a rotor with an external peripheral speed of 1 ft. per sec. are —

0.05 lb. per sq. in. tangential stress at the external radius
0.094 lb. per. sq. in. tangential stress at the internal radius and
0.007 lb. per. sq. in. maximum radial stress at 0.8 radius.

Since the stresses vary as the square of the external peripheral speed the stresses for an external peripheral speed of 400 ft. per sec. are —

$0.05 \times 400^2 = 8000$ lb. per sq. in. tangential stress at the external radius
$0.094 \times 400^2 = 15040$ lb. per sq. in. tangential stress at the internal radius
$0.007 \times 400^2 = 1120$ lb. per sq. in. maximum radial stress at 0.8 radius.

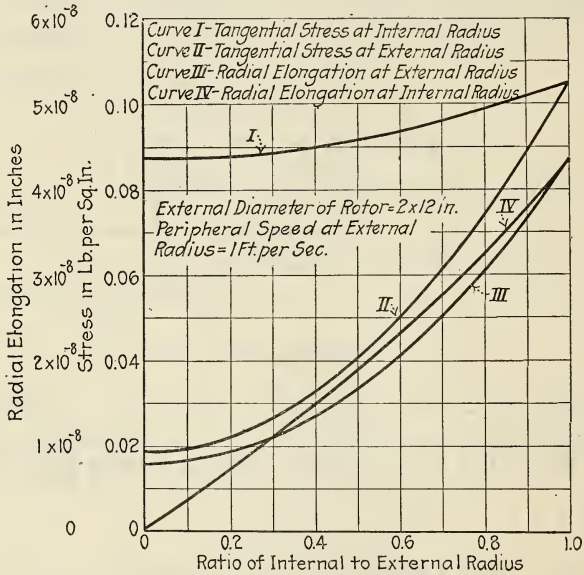

FIG. 11 STRESSES AND RADIAL ELONGATION OF HOLLOW CYLINDRICAL ROTOR

From Fig. 11 the total elongation in the radius for a rotor with a 12-in. radius and an external peripheral speed of 1 ft. per sec. is 2.075×10^{-8} in. Since the radial elongation is proportional to square of the external peripheral speed and the first power of the radius, the elongation in the radius of the rotor under consideration is $2.075 \times 10^{-8} \times 400^2 \times \frac{20}{12}$ or 0.005536 in., and the increase in the diameter is 2×0.005536 or 0.011072 in.

19 *Case III. Cylindrical Rotor with Central Hole and Externally Loaded* (Fig. 13). — This case applies to practically all rotors of both solid and laminated construction that have uniformly distributed windings imbedded in slots at the periphery.

20 Let —

S_{ro} = specific radial stress at external circumference due to external load

S_t = tangential stress across radial plane at distance x from axis

S_r = radial stress at distance x from neutral axis.

S_t and S_r include the loading-back effect of the external load which

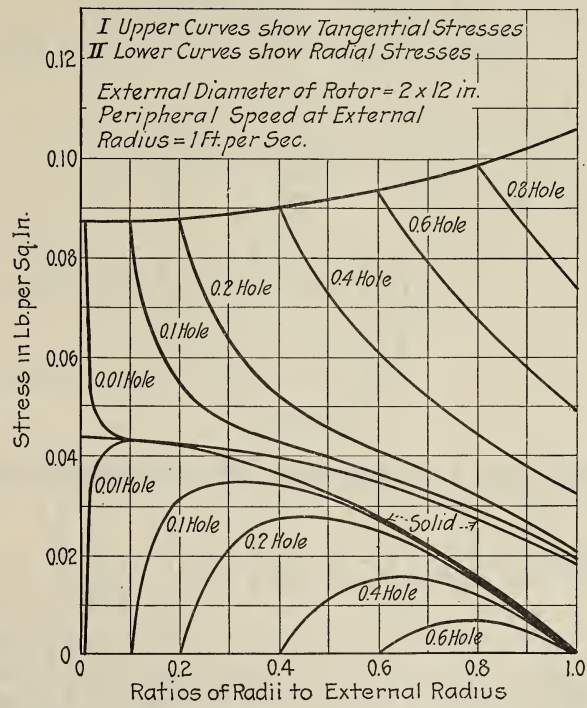

FIG. 12 RADIAL AND TANGENTIAL STRESSES OF HOLLOW CYLINDRICAL ROTOR

is transmitted to the elementary section. Hence, the general formulas derived for the rotating cylinder apply equally well in this case, when the proper constants of integration are determined.

21 The conditions in this case are —

$$\left.\begin{aligned} S_r &= S_{ro} \text{ when } x = R \\ S_r &= 0 \quad \text{when } x = r \end{aligned}\right\} \quad \cdots\cdots\cdots \quad [20]$$

Hence b_1 and b_2 can be determined by substituting these limiting conditions in Equation [10], which gives —

$$\left.\begin{aligned} \frac{1-v^2}{E} S_{ro} &= aR^2 (3+v) + b_1 (v+1) - \frac{b_2}{R^2} (1-v) \\ 0 &= ar^2 (3+v) + b_1 (v+1) - \frac{b_2}{r^2} (1-v) \end{aligned}\right\} \quad . \quad [21]$$

From which —

$$\left.\begin{aligned} b_1 &= -\frac{a(3+v)(r^2+R^2)}{1+v} + \frac{S_{ro}(1-v)R^2}{E(R^2-r^2)} \\ b_2 &= -\frac{a(3+v)r^2R^2}{1-v} + \frac{S_{ro}(1+v)r^2R^2}{E(R^2-r^2)} \end{aligned}\right\} \quad . \quad . \quad . \quad [22]$$

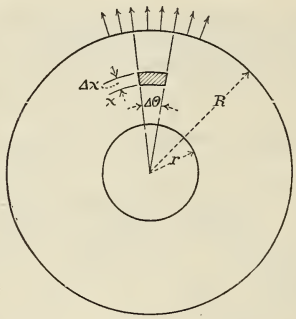

FIG. 13

Substituting b_1 and b_2 in Equations [10] and simplifying, gives —

$$S_r = 0.043606\, V_{fr}^2 \left[\frac{r^2}{R^2} + 1 - \frac{x^2}{R^2} - \frac{r^2}{x^2}\right] + S_{ro} \left[\frac{1 - \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right] \quad . \quad . \quad . \quad [23]$$

$$S_t = 0.043606\, V_{fr}^2 \left[\frac{r^2}{R^2} + 1 + \frac{r^2}{x^2} - 0.57575 \frac{x^2}{R^2}\right] + S_{ro} \left[\frac{1 + \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right] \quad [24]$$

When $x = R$,

$$S_t = 0.043606\, V_{fr}^2 \left[2 \frac{r^2}{R^2} + 0.42424\right] + S_{ro} \left[\frac{1 + \frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}}\right] \quad . \quad . \quad . \quad . \quad [25]$$

When $x = r$,

$$S_t = 0.043606\, V_{fr}^2 \left[2 + 0.042424 \frac{r^2}{R^2}\right] + 2S_{ro} \left[\frac{1}{1 - \frac{r^2}{R^2}}\right] \quad . \quad . \quad . \quad [26]$$

To find the radial elongation —

$$\lambda = ax^3 + b_1x + b_2x^{-1}$$

Substituting the values of b_1 and b_2 from Equation [22] and evaluating a gives —

$$\lambda_x = ax^3 - \frac{a(3+v)\ (r^2+R^2)x}{1+v} - \frac{a(3+v)r^2R^2}{x(1-v)} + \frac{S_{ro}(1-v)R^2x}{E(R^2-r^2)}$$

$$+ \frac{S_{ro}(1+v)r^2R^2}{E(R^2-r^2)x}$$

$$= 0.64 \times 10^{-9}\ V_{fr}{}^2x\left[1.65 + 1.65\frac{x^2}{R^2} - 0.65\frac{x^2}{R^2}\right.$$

$$\left. + 3.06\frac{r^2}{x^2}\right] + 34.5 \times 10^{-9}\ S_{ro}x\left[\frac{0.7+1.3\frac{r^2}{x^2}}{1-\frac{r^2}{R^2}}\right] \quad . \ . \ . \quad [27]$$

When $x = R$,

$$\lambda_R = 0.64 \times 10^{-9}\ V_{fr}{}^2R\left[1 + 4.71\frac{r^2}{R^2}\right]$$

$$+ 34.5 \times 10^{-9}\ S_{ro}R\left[\frac{0.7+1.3\frac{r^2}{R^2}}{1-\frac{r^2}{R^2}}\right] \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad [28]$$

$$\lambda_r = 0.64 \times 10^{-9}\ V_{fr}{}^2\,r\left[4.71 + \frac{r^2}{R^2}\right] + 34.5 \times 10^{-9}\ S_{ro}r\left[\frac{2}{1-\frac{r^2}{R^2}}\right] \quad [28a]$$

22 It is evident from the above equations that the effect of the external radial load is to add a variable term to the original equations for the stresses and radial elongation of hollow cylindrical rotor without an external load. The curves in Fig. 14 show the additional tangential stresses at the external and internal radii, and the total radial elongation of cylindrical rotors with different ratios of internal to external radii. The curves in Fig. 15 show the variation of the radial stress due to the external load for cylindrical rotors having different ratios of internal to external radii. The stresses are based on an external radial load of 1 lb. per sq. in. and the radial elongations are based on a rotor with a 12-in. external radius and an external load of 1 lb. per sq. in. Hence, in considering the same numerical example previously considered for the hollow cylindrical rotor, except that it has an uniform external radial load of 2000 lb. per sq. in. the additional terms are:

From the curves in Figs. 14 and 15,

2.13 lb. per sq. in. = tangential stress at the external radius
3.13 lb. per sq. in. = tangential stress at the internal radius
0.683 lb. per sq. in. = radial stress at 0.8 radius.

Since the stresses vary directly as the external radial load,

2000×2.13 = 4260 lb. per sq. in. = tangential stress at the external radius

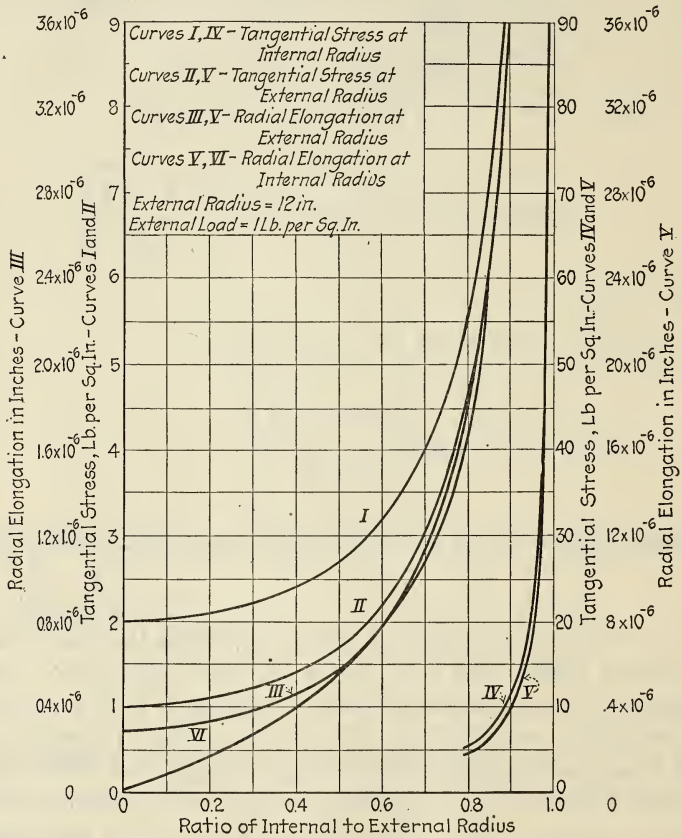

Fig. 14 Additional Tangential Stresses and Radial Elongation of a Hollow Cylindrical Rotor Due to External Load

2000×3.13 = 6260 lb. per sq. in. = tangential stress at the internal radius
2000×0.683 = 1366 lb. per sq. in. = radial stress at 0.8 radius.

Then the total stresses are —

$8000 + 4260 = 12260$ lb. per sq. in. = tangential stress at external radius
$15040 + 6260 = 21300$ lb. per sq. in. = tangential stress at internal radius
$1120 + 1366 = 2486$ lb. per sq. in. = radial stress at 0.8 radius.

23 From curve III of Fig. 14, the radial elongation of a rotor with a 12-in. external radius and an external radial load of 1 lb. per sq. in. is 0.76×10^{-6} in. Then the radial elongation for a rotor with a 20-in. radius and an external load of 2000 lb. per sq. in. is $0.76 \times 10^{-6} \times 2000 \times \frac{20}{12}$ or 0.002540 in. The total radial elongation at the external radius is 0.00254 + .005536 or 0.00808 in.

24 *Case IV. Solid Cylindrical Rotor without Central Hole but*

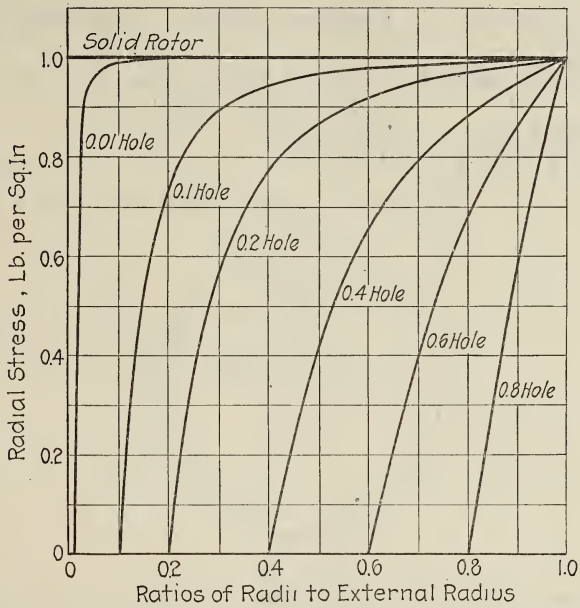

FIG. 15 VARIATION OF RADIAL STRESS DUE TO AN EXTERNAL LOAD FOR HOLLOW CYLINDRICAL ROTORS OF DIFFERENT RATIOS OF INTERNAL TO EXTERNAL DIAMETER

(External radial load = 1 lb. per sq. in.)

Externally Loaded (Fig. 16). — Refer to Fig. 13. In this case the limiting conditions are:

$$S_r = S_{ro} \text{ when } x = R$$

$$\lambda = 0 \text{ when } x = 0$$

since —

$$\lambda = ax^3 + b_1x + b_2x^{-1}$$

when —

$$x = 0, \lambda = 0. \quad \text{Hence } b_2 = 0$$

Substituting the condition that $S_r = S_{ro}$ when $x = R$ in Equation [10] gives —

$$\left.\begin{aligned} \frac{1 - v^2}{E} S_{ro} &= aR^2(3 + v) + b_1(v + 1) \\ \text{and} \qquad & \\ b_1(v + 1) &= \frac{1 - v^2}{E} S_{ro} - aR^2(3 + v) \end{aligned}\right\} \quad \ldots \quad [29]$$

Substituting $b_1(v + 1)$ in Equation [10] and simplifying, gives —

$$S_r = \frac{E}{1 - v^2}\left[ax^2(3 + v) - aR^2(3 + v)\right] + S_{ro}$$

$$= \frac{Ea(3 + v)}{1 - v^2}\left[x^2 - R^2\right] + S_{ro}$$

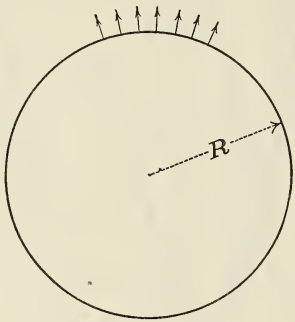

Fig. 16

$$= -0.043606\, V_{fr}^2\left[\frac{R^2}{x^2} - 1\right] + S_{ro}$$

$$= 0.043606\, V_{fr}^2\left[1 - \frac{x^2}{R^2}\right] + S_{ro} \text{ at any radius } x \quad \ldots \quad [30]$$

and —

$$S_t = 0.043606\, V_{fr}^2\left[1 - 0.57575\frac{x^2}{R^2}\right] + S_{ro} \text{ at any radius } x \quad [31]$$

when —

$$x = R$$
$$S_r = S_{ro}$$

and —

$$S_t = 0.043606\, V_{fr}^2 \times 0.42424 + S_{ro}$$

$$= 0.0185\, V_{fr}^2 + S_{ro} \quad \ldots \ldots \ldots \ldots \ldots \quad [32]$$

And when $x = 0$,

$$S_r = S_t = 0.043606\, V_{fr}^2 + S_{ro} \quad \text{[33]}$$

Since $b_1(v + 1) = \frac{1 - v^2}{E} S_{ro} - aR^2(3 + v)$

$$b_1 = \frac{1 - v}{E} S_{ro} - aR^2 \frac{3 + v}{1 + v}$$

$$b_2 = 0$$

therefore —

$$\lambda = ax^3 + b_1 x$$

$$= ax^3 + \left[\frac{(1 - v)S_{ro}}{E} - \frac{aR^2(3 + v)}{1 + v}\right] x$$

If $x = R$,

$$\lambda = 0.64 \times 10^{-9} \cdot V_{fr}^2 R + 24.1 \times 10^{-9}\, S_{ro} R \quad \text{[34]}$$

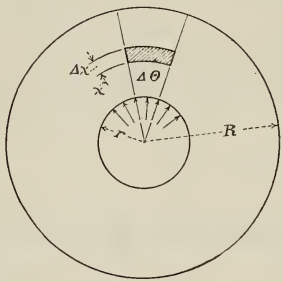

Fig. 17

25 Thus in the case of the solid rotor with an external radial load, the superimposed tangential and radial stresses at any fiber are constant and equal in magnitude to the radial stress at the outer radius due to the external load. If the solid rotor of the example previously considered has an external radial load of 2000 lb. per sq. in., the total stresses are —

6975 + 2000 = 8975 lb. per sq. in. tangential and radial stresses at the center of the rotor

2960 + 2000 = 4960 lb. per sq. in. tangential stress at the external radius.

26 The elongation in the radius due to the external load is $24.1 \times 10^{-9} \times 2000 \times 20 = 0.000964$ in., and the total radial elongation is 0.00205 + 0.000964 = 0.003014 in.

27 *Case V. Cylindrical Rotor with Central Hole and Internally Loaded* (Fig. 17). — A conspicuous example of this class is the coil retaining ring which supports the end turns of the field winding of a turbo rotor. The limiting conditions in this case are —

$$\left.\begin{aligned} S_r &= 0 \text{ when } x = R \\ S_r &= -S_{ro}' \text{ when } x = r \end{aligned}\right\} \quad \text{[35]}$$

where S_{ro}' = radial compressive stress at the internal radius. Substituting these conditions in Equation [10] gives —

$$\left.\begin{aligned} -\frac{1-v^2}{E}S_{ro}' &= ar^2(3+v) + b_1(1+v) - \frac{b_2}{r^2}(1-v) \\ 0 &= aR^2(3+v) + b_1(v+1) - (1-v)\frac{b_2}{R^2} \end{aligned}\right\} \quad [36]$$

28 Solving these simultaneous equations for b_1 and b_2 and substituting in Equation [10] gives —

$$S_t = 0.043606\, V_{fr}^2\left[\frac{r^2}{R^2} + \frac{r^2}{x^2} + 1 - 0.57575\frac{x^2}{R^2}\right] + S_{ro}'\left[\frac{\dfrac{r^2}{R^2} + \dfrac{r^2}{x^2}}{1 - \dfrac{r^2}{R^2}}\right] \quad [37]$$

$$S_r = 0.043606\, V_{fr}^2\left[\frac{r^2}{R^2} - \frac{r^2}{x^2} - \frac{x^2}{R^2} + 1\right] - S_{ro}'\left[\frac{\dfrac{r^2}{x^2} - \dfrac{r^2}{R^2}}{1 - \dfrac{r^2}{R^2}}\right] \quad [38]$$

When $x = R$, $S_r = 0$, and —

$$S_t = 0.043606\, V_{fr}^2\left[0.42424 + 2\frac{r^2}{R^2}\right] + 2S_{ro}'\frac{\dfrac{r^2}{R^2}}{1 - \dfrac{r^2}{R^2}} \quad [39]$$

When $x = r$, $S_r = -S_{ro}'$, and —

$$S_t = 0.043606\, V_{fr}^2\left[2 + 0.42424\frac{r^2}{R^2}\right] + S_{ro}'\left[\frac{1 + \dfrac{r^2}{R^2}}{1 - \dfrac{r^2}{R^2}}\right] \quad [40]$$

29 In order to get an expression for the radial elongation, Equation [36] can be solved for b_1 and b_2:

$$b_1 = \frac{S_{ro}'(1-v)\,r^2}{E\,(R^2 - r^2)} - \frac{a(3+v)\,(r^2 + R^2)}{1+v}$$

$$b_2 = \frac{S_{ro}'(1+v)\,R^2r^2}{E\,(R^2 - r^2)} - \frac{a(3+v)\,R^2r^2}{1-v}$$

Hence —

$$\begin{aligned} \lambda &= ax^3 + b_1x + b_2x^{-1} \\ &= ax^3 + \frac{S_{ro}'(1-v)r^2}{E\,(R^2 - r^2)}x - \frac{a(3+v)\,(r^2 + R^2)}{1+v}x \\ &\qquad + \frac{S_{ro}'(1+v)\,R^2r^2}{E\,(R^2 - r^2)\,x} - \frac{a(3+v)\,(R^2r^2)}{(1-v)\,x} \end{aligned}$$

$$\lambda = 0.64 \times 10^{-9} V_{fr}^2 x \left[1.65 + 1.65 \frac{r^2}{R^2} - 0.65 \frac{x^2}{R^2} + 3.06 \frac{r^2}{x^2} \right]$$

$$+ 34.5 \times 10^{-9} S_{ro}' x \left[\frac{0.7 \frac{r^2}{R^2} + 1.3 \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}} \right] \quad . \quad . \quad . \quad . \quad [41]$$

The total radial elongation at the external radius is—

$$\lambda = 0.64 \times 10^{-9} V_{fr}^2 R \left[1 + 4.71 \frac{r^2}{R^2} \right] + 69 \times 10^{-9} S_{ro}' R \left[\frac{\frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}} \right] [41a]$$

The total radial elongation at the internal radius is—

$$\lambda_r = 0.64 \times 10^{-9} \times V_{fr}^2 r \left[4.71 + \frac{r^2}{R^2} \right]$$

$$+ 34.5 \times 10^{-9} S_{ro}' r \left[\frac{1.3 + 0.7 \frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}} \right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [41b]$$

30 The curves in Fig. 18 show the additional tangential stresses at the internal and external radii, and the total radial elongation for hollow cylindrical rotors with an internal radial load of 1 lb. per sq. in. The variation of the radial stresses for different ratios of internal to external diameter are shown in Fig. 19. If the rotor in the example given under Case III has an internal radial load of 2000 lb. per sq. in. instead of the external radial load, the additional stresses and radial elongation would be as follows:

31 From the curves in Figs. 18 and 19 the additional stresses for an internal load of 1 lb. per sq. in. are —

2.125 lb. per sq. in. tangential stress at the internal radius
1.125 lb. per sq. in. tangential stress at the external radius
– 0.315 lb. per sq. in. radial stress (compression) at 0.8 radius
0.475×10^{-6} in. radial elongation for rotor with a 12-in. radius.

Then for an internal radial load of 2000 lb. per sq. in.,

$2000 \times 2.125 = 4250$ lb. per sq. in. tangential stress at the internal radius
$2000 \times 1.125 = 2250$ lb. per sq. in. tangential stress at the external radius
$-2000 \times 0.315 = -630$ lb. per sq. in. radial stress at 0.8 radius

$2000 \times 0.475 \times 10^{-6} \times \frac{20}{12} = 0.00158$ in. elongation in the radius.

32 Using the results of the problem in Case II the total stresses and radial elongation are —

15040 + 4250 = 19290 lb. per sq. in. total tangential stress at the internal radius
8000 + 2250 = 10250 lb. per sq. in. total tangential stress at the external radius
1120 − 630 = 490 lb. per sq. in. radial stress at 0.8 radius
0.005536 + 0.00158 = 0.00712 in. total radial elongation at the external radius.

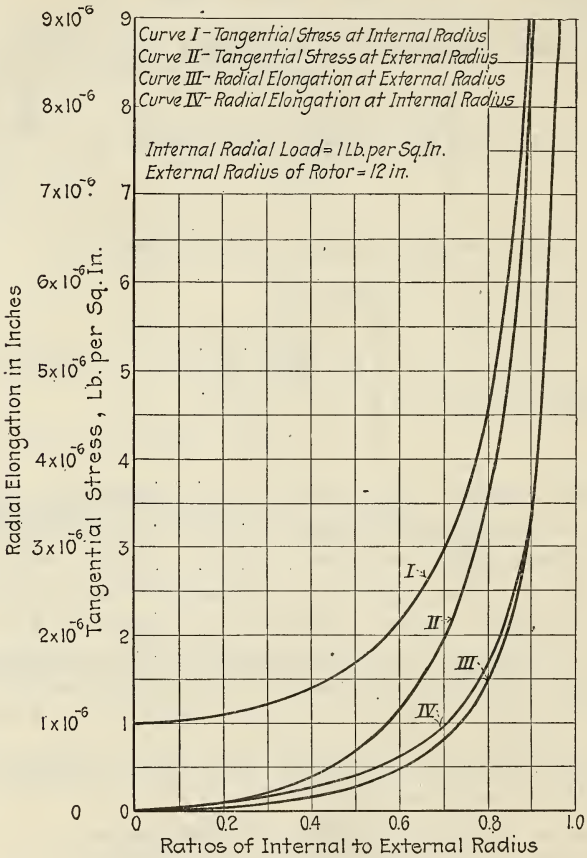

Fig. 18 Additional Tangential Stresses and Radial Elongation of Hollow Cylindrical Rotor with Uniform Internal Radial Load

33 *Case VI. Cylindrical Rotor with Central Hole and Loaded Internally and Externally* (Fig. 20). — Let S_{ro} be the radial stress at the outer circumference due to the external load, and $-S_{ro}'$ the radial stress at inner circumference due to internal load. The limiting conditions are —

$$S_r = S_{ro} \text{ when } x = R$$
$$S_r = -S_{ro}' \text{ when } x = r$$

34 Using these relations in Equation [10] gives

$$\left.\begin{aligned} \frac{1-v^2}{E} S_{ro} &= aR^2 (3+v) + b_1 (v+1) - \frac{b_2}{R^2}(1-v) \\ -\frac{1-v^2}{E} S_{ro}' &= ar^2 (3+v) + b_1 (v+1) - \frac{b_2}{r^2}(1-v) \end{aligned}\right\} \quad \ldots [42]$$

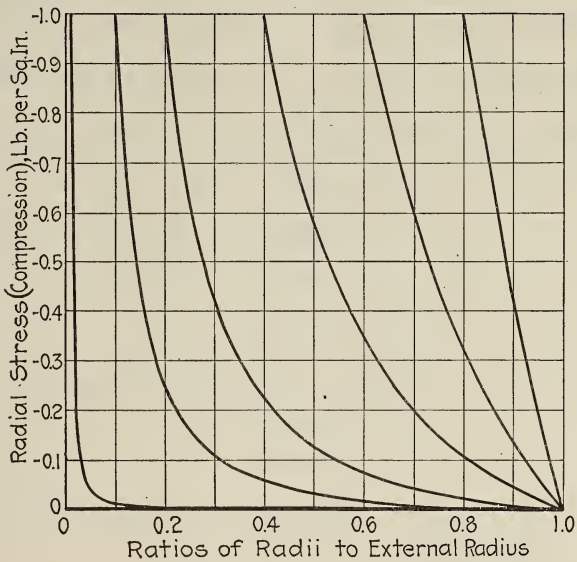

FIG. 19 VARIATION OF RADIAL STRESS DUE TO AN INTERNAL LOAD FOR HOLLOW CYLINDRICAL ROTORS HAVING DIFFERENT RATIOS OF INTERNAL TO EXTERNAL DIAMETER

(Internal radial load = 1 lb. per sq. in.)

Solving the two simultaneous Equations [42] for b_1 and b_2 gives—

$$\left.\begin{aligned} b_1 &= \frac{(1-v)\,(S_{ro}R^2 + S_{ro}'r^2)}{E\,(R^2 - r^2)} - \frac{a(3+v)\,(R^2+r^2)}{1+v} \\ b_2 &= \frac{(1+v)\,(S_{ro} + S'_{ro})\,R^2\,r^2}{E\,(R^2 - r^2)} - \frac{a(3+v)\,R^2\,r^2}{1-v} \end{aligned}\right\} \quad \ldots [43]$$

Substituting these values of b_1 and b_2 in Equations [10] gives the radial and tangential stresses at any fiber.

$$S_r = 0.043606 V_{fr}^2 \left[\frac{r^2}{R^2} + 1 - \frac{x^2}{R^2} - \frac{r^2}{x^2}\right] + S_{ro} \left[\frac{1 - \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right]$$

$$+ S_{ro}' \left[\frac{\frac{r^2}{R^2} - \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right] \quad \ldots \ldots \ldots \ldots \quad [44]$$

$$S_t = 0.043606 V_{fr}^2 \left[\frac{r^2}{R^2} + 1 + \frac{r^2}{x^2} - 0.57575 \frac{x^2}{R^2}\right] + S_{ro} \left[\frac{1 + \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right]$$

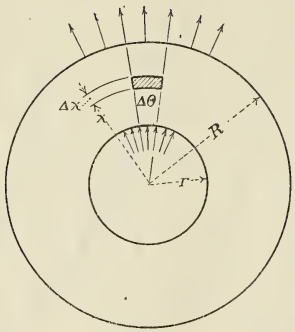

Fig. 20

$$+ S_{ro}' \left[\frac{\frac{r^2}{R^2} + \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right] \quad \ldots \ldots \ldots \ldots \quad [45]$$

35 The stresses at the external radius are —

$$S_r = S_{ro} \quad \ldots \ldots \ldots \quad [46]$$

$$S_t = 0.043606 V_{fr}^2 \left[0.42424 + 2\frac{r^2}{R^2}\right] + S_{ro} \left[\frac{1 + \frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}}\right]$$

$$+ 2 S_{ro}' \left[\frac{\frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}}\right] \quad \ldots \ldots \ldots \ldots \quad [47]$$

36 Similarly, the values of the stresses at the internal radius are —

$$S_r = -S_{ro}' \quad \text{[48]}$$

$$S_t = 0.043606 V_{fr}^2 \left[2 + 0.42424 \frac{r^2}{R^2}\right] + 2 S_{ro} \left[\frac{1}{1 - \frac{r^2}{R^2}}\right] + S_{ro}' \left[\frac{1 + \frac{r_2}{R^2}}{1 - \frac{r^2}{R^2}}\right] \quad \text{[49]}$$

37 The radial elongation at any fiber, from Equation [9], is —

$$\lambda_x = 0.64 \times 10^{-9} V_{fr}^2 x \left[1.65 + 1.65 \frac{r^2}{R^2} - 0.65 \frac{x^2}{R^2} + 3.06 \frac{r^2}{x^2}\right] + 34.5 \times 10^{-9} S_{ro} x \left[\frac{0.7 + 1.3 \frac{r^2}{x}}{1 - \frac{r^2}{R^2}}\right] + 34.5 \times 10^{-9} S_{ro}' x \left[\frac{0.7 \frac{r^2}{R^2} + 1.3 \frac{r^2}{x^2}}{1 - \frac{r^2}{R^2}}\right] \quad \text{[50]}$$

When $x = R$,

$$\lambda_R = 0.64 \times 10^{-9} V_{fr}^2 R \left[1.0 + 4.71 \frac{r^2}{R^2}\right] + 34.5 \times 10^{-9} S_{ro} R \left[\frac{0.7 + 1.3 \frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}}\right] + 69 \times 10^{-9} S_{ro}' R \left[\frac{\frac{r^2}{R^2}}{1 - \frac{r^2}{R^2}}\right] \quad \text{[51]}$$

When $x = r$,

$$\lambda_r = 0.64 \times 10^{-9} V_{fr}^2 r \left[4.71 + \frac{r^2}{R^2}\right] + 34.5 \times 10^{-9} S_{ro} r \left[\frac{2}{1 - \frac{r^2}{R^2}}\right]$$

$$+ 34.5 \times 10^{-9} S_{r_o}' r \left[\frac{1.3 + 0.7 \dfrac{r^2}{R^2}}{1 - \dfrac{r^2}{R^2}} \right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [51a]$$

38 It is evident from Equations [44] to [51] that the additional stresses and radial elongation due to internal and external radial loads applied simultaneously are equal in value to the sum of the corresponding quantities when the loads are applied separately. Hence, by using the curves in Figs. 14, 15, 18 and 19, the total additional stresses and elongation can be determined for a hollow cylindrical rotor with any internal and external radial loads. If the rotor given in the problem of Case II has an internal and external radial load of 2000 lb. per sq. in., the total values, stresses and elongation from Cases III and V, are —

15040 + 6260 + 4250 = 25550 lb. per sq. in. tangential stress at the internal radius
8000 + 4260 + 2250 = 14510 lb. per sq. in. tangential stress at the external radius
1120 + 1366 − 630 = 1850 lb. per sq. in. radial stress at 0.8 radius
0.005536 + 0.00254 + 0.00158 = 0.00966 in. elongation in the external radius.

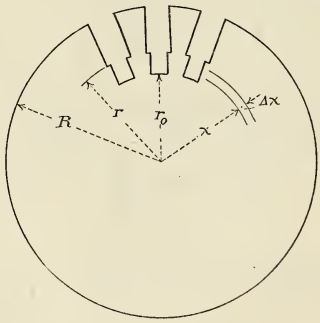

Fig. 21

39 *Case VII. Calculation of Stresses in Rotor Teeth* (Fig. 21). — In this problem it is assumed that the equivalent weight per cubic inch of the materials in the slots is the same as that of the iron and that the end turns are supported independently.

40 Let r = radius at base of slots
R = radius at top of slots
w = angular velocity in radians per sec.
W = weight per cu. in. of material
p = ratio of tooth width at the bottom of the slots to the circumference at this radius
S_r = radial stress at the root of the teeth

Consider an annular ring at a distance x from the center and radial depth Δx. Then —

$$\Delta V = Kx \cdot \Delta x = \text{volume of elementary ring}$$

$$\Delta M = K\frac{W}{g}\, x \cdot \Delta x = \text{mass of elementary section}$$

$$\Delta F = \frac{W}{12g}\, Kx^2\, w^2 \cdot \Delta x = \text{centrifugal force acting on ring}$$

$$F = \frac{W}{12g}\, Kw^2 \int_r^R x^2\, dx = \text{total centrifugal force of material between radii } r \text{ and } R$$

$$= \frac{W}{12g}\, Kw^2\, \frac{R^3 - r^3}{3}$$

41 This total force F must be carried by the iron section at the root of the teeth.

Krp = iron section at the root of the teeth, and
$rpKS_r$ = total force acting on the material at the root of the teeth

$$rpKS_r = \frac{W}{12g}\, Kw^2\, \frac{R^3 - r^3}{3}$$

$$pS_r = \frac{W}{12g}\, \frac{w^2}{3r}\, (R^3 - r^3)$$

$$= \frac{W}{36g}\, w^2\, \frac{R^3}{r} \left(1 - \frac{r^3}{R^3}\right)$$

$$= \frac{4W}{g}\, V_{fr}^2 \left(\frac{R}{r} - \frac{r^2}{R^2}\right)$$

42 If the maximum permissible stress at the root of the teeth is S_r', then —

$$p = \frac{0.0354}{S_r'}\, V_{fr}^2 \left[\frac{R}{r} - \frac{r^2}{R^2}\right] \quad \ldots\ldots \quad [52]$$

or for a given tooth section at the base of the teeth,

$$S_r = \frac{0.0354}{p}\, V_{fr}^2 \left[\frac{R}{r} - \frac{r^2}{R^2}\right] \quad \ldots\ldots \quad [53]$$

Numerical Example:

If $p = 0.5$, $\frac{r}{R} = 0.8$, and $V_{fr} = 400$ ft. per sec., then —

$$S_r = \frac{0.0354}{0.5} \times (400)^2\, (1.25 - 0.64)$$

$$= 6900 \text{ lb. per sq. in. radial stress at the root of the teeth.}$$

43 *Case VIII. Stresses in Teeth at Bottom of Air Ducts* (Fig. 22). — In the rotors of radial-slot turbo-alternators air channels are often provided at the bottom of the slots for cooling purposes. The iron section at the bottom of the channels must carry the total centrifugal force due to all of the external material.

44 Let R = external radius of the rotor

r = radius at the bottom of main slots

r_o = radius at the bottom of the air channel

p_1 = ratio of the tooth width at the bottom of air channels to the circumference at this point

S_r = lb. per sq. in. stress in iron at the bottom of the teeth.

Then—

$$K(1 - p_1)\, r_o = \text{total width of the air channels}$$

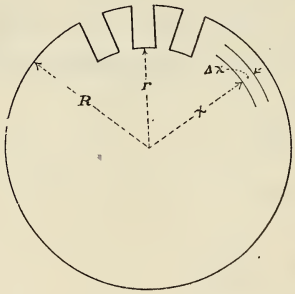

Fig. 22

and—

$$K[x - (1 - p_1)\, r_o] = \text{total iron width at any section}$$

$$\Delta V = K\,[x - (1 - p_1)\, r_o]\, \Delta x = \text{volume of elementary section}$$

$$\Delta M = K\,\frac{W}{g}\,[x - (1 - p_1) r_o]\, \Delta x = \text{mass of elementary section}$$

$$\Delta F_o = \Delta M \cdot \frac{x}{12}\, w^2 = \text{centrifugal force acting on elementary section}$$

$$= K\,\frac{W}{12g}\, w^2\, [x^2 - (1 - p_1) r_o x]\, \Delta x$$

$$F_o = K\,\frac{W}{12g}\, w^2 \int_{r_o}^{r} \left[x^2 - (1 - p_1)\, r_o x\right] dx$$

$$= K\frac{W}{12g}w^2\left[\frac{x^3 - r_o^3}{3} - \frac{1 - p_1}{2}(r^2 - r_o^2)\right]$$

45 The total centrifugal force acting on the section at the bottom of the air channel is $F + F_o$, where F is determined as in the previous case.

$$F + F_o = K\frac{W}{12g}w^2\left[\frac{R^3 - r_o^3}{3} + \frac{1 - p_1}{2}(r_o^3 - r_o r^2)\right]$$

But $r_o p_1 K S_r$ = total force acting on the teeth at the bottom of the air channel, therefore —

$$r_o p_1 K S_r = \frac{W}{12g}Kw^2\left[\frac{R^3 - r_o^3}{3} + \frac{1 - p_1}{2}(r_o^3 - r_o r^2)\right]$$

$$p_1 S_r = \frac{Ww^2}{12gr_o}\left[\frac{R^3 - r_o^3}{3} + \frac{r_o^3 - r_o r^2}{2}\right] - \frac{Ww^2 p_1}{12gr_o^2}\left[r_o^3 - r_o r^2\right]$$

$$p_1\left[S_r - \frac{Ww^2}{12gr_o^2}(r_o r^2 - r_o^3)\right] = \frac{Ww^2}{12gr_o}\left[\frac{R^3}{3} + \frac{r_o^3}{6} - \frac{r_o r^2}{2}\right]$$

$$S_r = \frac{0.1061}{p_1}V_{fr}^2\left[\frac{R}{3r_o} + \frac{r_o^2}{6R^2} - \frac{r^2}{2R^2} - \frac{p_1}{2}\left(\frac{r_o^2}{R^2} - \frac{r^2}{R^2}\right)\right] \quad . \quad . \quad [54]$$

or —

$$p_1 = \frac{0.1061V_{fr}^2\left[\frac{R}{3r_o} + \frac{r_o^2}{6R^2} - \frac{r^2}{2R^2}\right]}{S_r + 0.0531V_{fr}^2\left[\frac{r_o^2}{R^2} - \frac{r^2}{R^2}\right]} \quad . \quad . \quad . \quad . \quad [55]$$

Numerical Example:

If $p_1 = 0.50$, $V_{fr} = 400$ ft. per sec., $\frac{r}{R} = 0.8$, and $\frac{r_o}{R} = 0.6$, then —

$$S_r = 0.1061 \times \frac{(400)^2}{0.5}\left[\frac{1}{3 \times 0.6} + \frac{(0.6)}{6} - \frac{(0.8)^2}{2} - \frac{0.5}{2}(0.6^2 - 0.8^2)\right]$$

= 12,400 lb. per sq. in. at the bottom of the air channels.

DISCUSSION

G. M. Eaton. This paper places a very valuable set of tools in the hands of designers of apparatus for high rotative speed. The text is very clear in its presentation of the fundamental principles, and the equations as finally derived are in practical and usable form.

In common with all formulas derived by pure analytic methods

the application in actual designs must be guided by a full knowledge of the materials to be employed, and the conditions to be met in service.

This may be well illustrated by one of the problems facing the designer of a rotor that is to be made from a single forging. On the face of the formulas there is no possible argument for an axial hole through the rotor, as the stresses at the surface of the hole are far higher than those existing at the axis in a similar rotor with a solid center. But after all, we are more interested in the factor of safety than we are in stress, and the question forces itself upon us, whether in this large forging the center actually is solid. The steel mills tell us that even when the cropping of the ingot has removed all appearance of piping, we must expect an area of weak structure extending for an indeterminate distance below the pipe. Furthermore, comparatively small temperature inequalities over the length of the ingot combine with the unavoidable temperature gradients normal to all radiating surfaces, to set up severe and sometimes destructive tensile stresses in the last metal to cool; namely, the axial core. The steel mills, therefore, are practically unanimous in recommending an axial inspection hole throughout the entire length of large forgings for high-speed rotors.

Apparently then, we must choose between a comparatively low calculated stress with quite indeterminate material on the one hand, and a high stress with material whose characteristics are in a measure capable of determination. This latter stress we may find leaves a factor of safety which appears at first thought to be lower than we would like to provide, when compared with the known elastic and endurance limits of the material employed.

From the standpoint of endurance limit, it is usually comforting to realize the infrequency with which the stress cycle from zero speed to full speed is completed by large high-speed rotating machinery.

There is, however, a further consideration in connection with normal speed stresses which constitutes a strong argument in favor of providing the axial hole. If the stress under discussion is close to the elastic limit at normal speed, it is entirely possible that under the test or occasional service over-speed condition, the rapidly mounting stress will pass the elastic limit. This is true whether a hole is intentionally provided, or whether an incipient weakness or rupture exists in a supposedly solid rotor. In the former case the material at the smooth surface of the hole will take a fairly uniform

permanent set, while in the latter a running crack may possibly develop.

Analyzing the effect of plastic flow of the metal adjacent to the hole, we find that since there can be no reduction of the resultants of centrifugal action at any given speed, and since the material undergoing plastic flow can offer little more than its elastic resistance to these resultants, the stress must be shared by material other than that undergoing plastic flow. Obviously, a loading back occurs and the less strained material adjacent to that which first flows will take on added stress. Some of this material will also enter the stage of plastic flow, the amplitude of such flow gradually decreasing with increase of radius and this increase being with no sharp line of demarcation, until a region is reached where the stress is exactly equal to the elastic limit. Outside of this region all phenomena remain elastic.

When the period of over-speed is passed and the speed begins to drop, the material that has been stretched beyond the elastic limit is too long to reassume the identical relations that it previously bore to adjacent material, and it enjoys a relief from the stress it originally endured at normal speed, this relief being approximated by the distance of plastic flow per unit of length as measured on the elastic scale of the material.

Therefore, a machine which has successfully passed an over-speed test will frequently run at normal speed with a lower maximum stress than occurred before the machine was over-speeded, and therefore with a greater elastic factor of safety than would appear from the formulas given in the paper.

We have referred to elastic factor of safety. This term is applied to the ratio of the elastic limit to the working stress. But in the case under consideration this ratio is quite misleading, and the actual factor of safety may be many times as great as this ratio would make it appear.

When we pull a standard test piece, we find that our material will undergo a certain per cent of elongation between gage points before it breaks. This elongation is not evenly distributed over the distance between gage points and is therefore not a fully representative characteristic of the material, but it is, nevertheless, somewhat indicative. The reduction of area is a much more definite description of the material.

We can pull a large number of test pieces and we find quite consistent results. Now if we base our factor of safety on the

elongation or reduction of area of the metal undergoing plastic flow, we have a more truly descriptive factor of safety which we may term the plastic factor of safety. The test piece will never break with average material till a certain reduction has occurred, and if the movement is positively stopped at a certain per cent of the available ultimate reduction, the ratio of the ultimate to the actual reduction is a logical safety factor. Numerically, we would never accept as low a plastic factor as the usual elastic factor of safety, but as a real and usable factor, the plastic factor of safety must be accepted in cases of the type under discussion.

The fundamental point of difference between the usual elastic beam condition and the setting of combined elastic and plastic phenomena is that in the former case the material follows the force application without relief of stress, under the laws of elasticity, whereas in the latter there occurs, as stated, a loading back of stress to lower stressed parts so that more and more material is loaded as the material in the critical region refuses to assume the share of the load it previously bore.

There is therefore no cause for anxiety in cases of combined elastic and plastic flow over what appears to be a low elastic factor of safety when further investigation along the lines suggested reveals an ample plastic factor of safety.

William Knight. The remarks made by Mr. Laffoon about the desirability of having a more comprehensive analysis of stresses in rotating elements of electrical machines than is given in handbooks are quite to the point.

As far, however, as stresses in rotating disks with a hole at the center are concerned, the writer believes that the analysis of such stresses, as it is given in Stodola's book, The Steam Turbine, is quite complete and comprehensive.

The only objection to the formulas given by Stodola, in my estimation, is that they involve an elaborate mathematical manipulation of numerical values and too many chances of making errors.

To eliminate, as far as possible, chances of errors in the calculation of stresses in rotating disks with a hole at the center and to make such calculation accessible to any ordinary draftsman, the writer has simplified the formulas given by Stodola and has expressed them in such a form as to make it possible to substitute the use of graphical methods for the long calculations involved in the use of them.

The two general equations given by Stodola for disks are as follows:

$$\sigma_r = \frac{E}{1 - v^2}\left[(3 + v)kx^2 + (\psi_1 + v)b_1x^{\psi_1 - 1} + (\psi_2 + v)b_2x^{\psi_2 - 1}\right] \quad . \quad . \quad [56]$$

$$\sigma_t = \frac{E}{1 - v^2}\left[(1 + 3v)kx^2 + (1 + \psi_1 v)b_1x^{\psi_1 - 1} + (1 + \psi_2 v)b_2x^{\psi_2 - 1}\right] \quad [57]$$

and refer to disks with a hole in the center having a profile represented by the equation:

$$y = \frac{c}{x^a}$$

where y is the thickness of the disk at any radius x, c is a constant, and a may be either positive or negative or equal zero.

For $a = 0$ we have the case of a disk of uniform thickness.

For positive values of a we have the disk section used by de Laval for small wheels.

For negative values of a we have a profile of disk increasing in thickness from the bore toward the periphery.

The exponent a is given by:

$$a = \frac{\log \dfrac{y_1}{y_2}}{\log \dfrac{x_2}{x_1}}$$

where y_1 and y_2 are the thicknesses of the disk at the corresponding radii x_1 and x_2.

b_1 and b_2 in the above formulas are two constants depending on the outline of the disk and on the presence or absence of other forces acting on the disk besides the centrifugal force of its own mass.

$$\psi_1 = \frac{a}{2} + \sqrt{\frac{a^2}{4} + av + 1}$$

$$\psi_2 = \frac{a}{2} - \sqrt{\frac{a^2}{4} + av + 1}$$

v = Poisson's ratio (for steel $v = 0.3$)

and, with the inch-lb-second system:

ω = angular velocity in radians per sec.
μ = unit of mass of metal of disk (steel = 000,725)
E = modulus of elasticity (steel = 30,000,000)

$$k = \frac{-(1 - v^2)\,\mu\,\omega^2}{E\left[8 - (3 + v)a\right]}$$

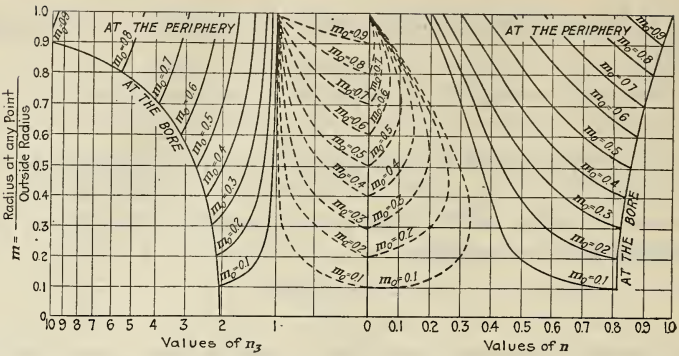

FIG. 23 STEEL DISK OF UNIFORM SECTION ($\alpha = 0$) WITH A CENTRAL HOLE

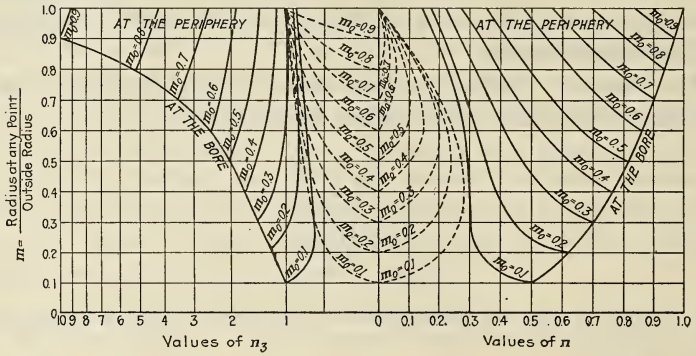

FIG. 24 STEEL DISK OF HYPERBOLIC SECTION ($\alpha = 0.5$) WITH A CENTRAL HOLE

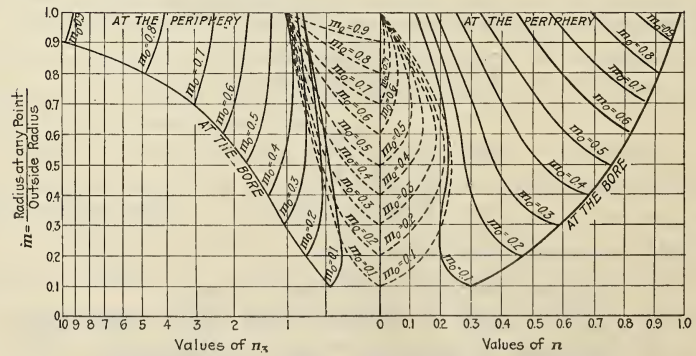

FIG. 25 STEEL DISK OF HYPERBOLIC SECTION ($\alpha = 1$) WITH A CENTRAL HOLE

FIGS. 23 TO 25 VALUES OF n

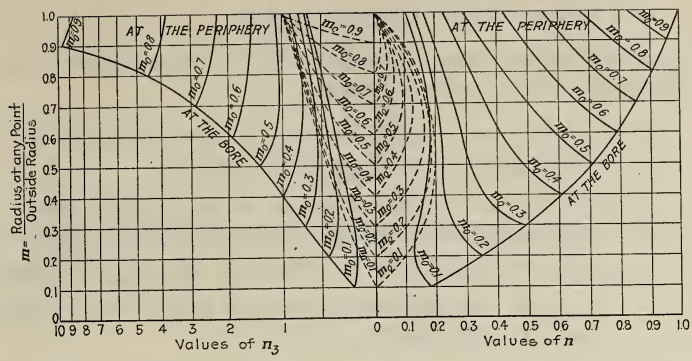

FIG. 26 STEEL DISK OF HYPERBOLIC SECTION ($\alpha = 1.5$) WITH A CENTRAL HOLE

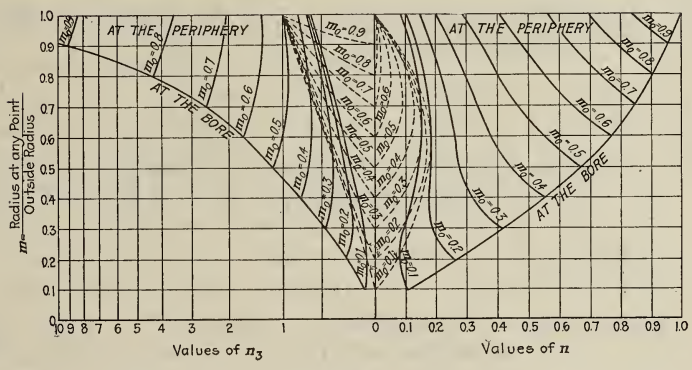

FIG. 27 STEEL DISK OF HYPERBOLIC SECTION ($\alpha = 2$) WITH A CENTRAL HOLE

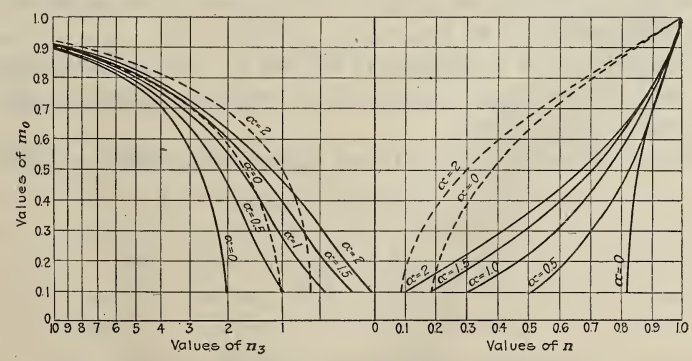

FIG. 28 STEEL DISK OF HYPERBOLIC SECTION

FIGS. 26 TO 28 VALUES OF n

x = radius at any point in inches.

In the simplified form in which Stodola's formulas are expressed, the following symbols will be used:

σ_t, σ_r = tangential and radial stresses at any radius due to the centrifugal forces of the disk only

σ_t^1, σ_r^1 = tangential and radial stresses at any radius due to the centrifugal force of the projecting masses only

σ = tangential stress in a thin ring of the same material as the disk, and rotating at a speed equal to the peripheral speed of the disk

m = ratio of radius at any point to the outside radius

m_0 = ratio of radius at the bore to the outside radius.

By introducing the two terms m and m_0, Equations [**56**] and [**57**] can be expressed as follows:

$$\sigma_r = \sigma \left\{ A \left[-m^2 + \frac{m^{\psi_1-\psi_2}(m_0^{3-\psi_2} - 1) - (m_0^{3-\psi_2} - m_0^{\psi_1-\psi_2})}{m^{1-\psi_2}(m_0^{\psi_1-\psi_2} - 1)} \right] \right\} = \sigma n \quad . \quad [\mathbf{58}]$$

$$\sigma_t = \sigma \left\{ B \left[-m^2 + \frac{Cm^{\psi_1-\psi_2}(m_0^{3-\psi_2} - 1) + D(m_0^{3-\psi_2} - m_0^{\psi_1-\psi_2})}{m^{1-\psi_2}(m_0^{\psi_1-\psi_2} - 1)} \right] \right\} = \sigma n \quad . \quad [\mathbf{59}]$$

The values of n in Equation [**58**] are obtained on the right hand side of the diagrams, Figs. 23 to 28, from dotted line curves for corresponding values of m and m_0.

The values of n in Equation [**59**] are also obtained on the right hand side of the same diagrams, but they are given by the full line curves there shown.

The constants A, B, C, D used in Equations [**58**] and [**59**] are:

$$A = (2 - v^2)(1 - v^2)B$$

$$B = \frac{2 - v^2}{8 - (3 + v)a}$$

$$C = -(2 - v^2)(1 - v^2)\psi_2$$

$$D = (2 - v^2)(1 - v^2)\psi_1$$

which for $v = 0.3$ become:

$$A = 1.74B$$

$$B = \frac{1.91}{8 - 3.3\alpha}$$

$$C = -1.74\psi_2$$

$$D = 1.74\psi_1$$

Similarly, introducing again the two terms m and m_0, Equations [56] and [57] can be expressed in this simplified form:

$$\sigma_r^1 = \sigma_3 \frac{m_0^{\psi_1-\psi_2} - m^{\psi_1-\psi_2}}{m^{1-\psi_2}(m_0^{\psi_1-\psi_2} - 1)} = \sigma_3 n_3 \quad . \quad . \quad . \quad . \quad [60]$$

$$\sigma_t^1 = \sigma_3 \frac{\psi_1 m_0^{\psi_1-\psi_2} - \psi_2 m^{\psi_1-\psi_2}}{m^{1-\psi_2}(1 - m_0^{\psi_1-\psi_2})} = \sigma_3 n_3 \quad . \quad . \quad . \quad [61]$$

where σ_3 = centrifugal force of projecting masses per square inch of surface at the periphery of the disk.

Values of n_3 in Equations [60] and [61] are obtained on the left hand side of diagrams Figs. 23 to 28, from dotted line curves for Equation [60], and from full line curves for Equation [61].

The usefulness of these diagrams is quite evident. In order to find the radial and tangential stresses at any point of a disk with a hole at the center, either of a uniform section (same as it is in the case considered by Mr. Laffoon) or of a hyperbolic section (which includes the case of the disk of uniform section), same as considered by Stodola, all we need to know is the value of α (which in the case considered by Mr. Laffoon is equal to zero), the values of m and m_0 and calculate on the slide rule the values of σ and σ_3. The diagrams will give us at a glance the values of radial and tangential stresses at any point between the bore and the periphery of the disk.

Diagrams Figs. 23 to 27 have been calculated for values of α varying from $\alpha = 0$ to $\alpha = 2$ ($\alpha = 0$, $\alpha = 0.5$, $\alpha = 1.0$, $\alpha = 1.5$, $\alpha = 2$) and for values of m and m_0 varying from 0.1 to 1.0. The diagram shown in Fig. 28 gives the values of tangential stress only at the bore and at the periphery of the disk for corresponding values of α and m_0. This diagram will be found to be particularly useful in determining the influence of a change in the design of the disk over its strength.

In order to allow anybody to plot these diagrams, the values of $\frac{\sigma_r}{\sigma}$ corresponding to the values of n in Equation [58] may be

taken from Tables 6 to 10, p. 517, *Machinery*, February 1918, which I published some years ago.

Tables 1 to 5 on the same page give the values of $\frac{\sigma_t}{\sigma}$ corresponding to the values of n in Equation [**59**].

Tables 16 to 20 on p. 518, February 1918 issue of *Machinery*, give the values of $\frac{\sigma_r^1}{\sigma_3}$ corresponding to the values of n_3 in Equation [**60**].

Tables 11 to 15 on the same page give the values of $\frac{\sigma_t^1}{\sigma_3}$ corresponding to the values of n_3 in Equation [**61**].

Figs. 23 to 28 were published by the writer in *Engineering* (London), August 3, 1917.

The particular case of the design of the rotor core of an electrical machine, considered by Mr. Laffoon in his paper, the writer has given in the *Electrical World*, January 12, 1918.

A. L. Kimball. The subject of stress around a hole has been investigated by an interesting method. A rubber diaphragm was ruled with squares, and by means of the distortion of the squares around the hole, an approximate value of the stress produced at the hole was arrived at. The stress did not depend on the size of the hole. The rubber must not be strained too much, or an error will result.

In Dr. Coker's laboratory in London some stressed celluloid sheets with elliptical holes were studied photo-elastically. It was found that the maximum stress increased as the ellipse was elongated and the amount of the stress was found to check almost exactly with the theoretical value. Compression at the top and bottom of the ellipse was discovered as the theory predicted. This can be used to show what occurs at a crack or notch, which may be considered a very elongated ellipse.

Paul Heymans. The mathematical treatment of problems of the kind which have been considered in Mr. Laffoon's paper can only be carried out at the price of simplifying assumptions. How much these assumptions cause the calculated stresses to depart from the actual stresses which will arise in the structure is always an uncertain question. The factor of safety has to cover this factor of doubt. We venture to say that the photo-elastic method opens a promising possibility for the investigation of the majority of

these problems. In this method the stresses are directly measured. The only restricting assumptions are that the material is isotropic and that it obeys Hooke's law of linear proportionality between stress and strain.

The stress analysis in the photo-elastic method is made on celluloid models. For those materials which have beyond the elastic limit a stress-strain curve similar to that of celluloid, the results obtained above that limit, where the material no more obeys Hooke's law but follows some other stress-strain curve, still hold. The stress-strain curve of celluloid is similar to the stress-strain curve of mild steel. Moreover, it can be acted upon by proper nitration of the nitro-cellulose entering in the compositions of celluloid. Therefore, a model of proper celluloid analysed below and beyond the elastic limit should throw fuller light on the subject treated in this paper.

The question of the stresses set up by a circular hole drilled in the center of a rotating disk has been raised. An Austrian physicist, Dr. Leon,[1] has presented a complete analytical treatment of the subject of stresses around a hole, and has shown that in the case of a plate uniformly stressed in all directions, a hole drilled in the plate will present around the boundary a stress double the previous uniform stress. In the case of an infinite plate uniformly stressed in one direction, this same discontinuity will set up stresses equal to three times the mean stress.

We have in our photo-elastic laboratory subjected a disk of celluloid to a uniform stress, and re-examined it after drilling in its center a small hole. The values of the stresses determined confirmed Dr. Leon's analytical results.

S. A. Moss. A rotor with a hole in the middle is certainly stressed much more than it would be if there were no hole. The radial stresses on either side of the hole do not balance each other as they do where there is no hole. A simple illustration is that of the effect produced by sticking a pin through a thin sheet of stretched rubber. The pin hole instantly enlarges. The conditions are the same at the center of the wheel.

Alphonse A. Adler said that everyone who analyzed such a subject as this commenced with equations of the maximum strain theory. He would like some one to base the equation on maximum shear

[1] Dr. A. Leon, Oesterreichische Wochenschrift für den öffenlichen Baudienst Wien, Februar 1908, pp. 163–168.

theory since this has recently become very prominent. He thought that the photo-elastic method would be valuable in determining whether to use the maximum strain or the maximum shear theory.

The author follows the usual custom of assuming that in the case of rotor teeth the stress in the main body of the rotor is only affected by the increased loading due to the teeth. From the paper by Heymans and Kimball[1] on Railway Motor Pinions it is shown how seriously the presence of teeth on the gear affects the distribution of stress in the hub. Perhaps this same influence is present in the case of armatures with toothed cores. A photo-elastic experiment would do much to indicate the departure from the simple assumptions given in this paper.

[1] See paper No. 1859.

No. 1859

STRESS DISTRIBUTION IN ELECTRIC-RAILWAY MOTOR PINIONS AS DETERMINED BY THE PHOTO-ELASTIC METHOD

By Paul Heymans, Cambridge, Mass.
Associate-Member of the Society
and
A. L. Kimball, Jr.,[1] Schenectady, N. Y.
Non-Member

This paper embodies some results of a general scientific study undertaken by the General Electric Company for the development of superior electric-railway motor pinions. The particular portion of the work described was performed at the Massachusetts Institute of Technology, using the General Electric Company's apparatus for stress determination in transparent models by the photo-elastic method. Some of the supplementary mechanical tests were made at Schenectady, and throughout the work close contact was maintained with the Railway Motor Department and the Research Laboratory at Schenectady. A brief description and discussion of the photo-elastic method is given in the first part. The stress distribution in, and the causes of ruptures of, given types of gear pinions used in electric-railway motors, as investigated by the photo-elastic method, are afterward reported upon and discussed.

I — DESCRIPTION OF THE METHOD

HOW TO DEFINE THE STATE OF STRESS AT ANY POINT OF A SOLID BODY

THE state of stress at any point in a solid body is determined when the traction across every plane through the point is known. There exist at any point three orthogonal planes across which the traction is purely normal and which are called the planes of principal stress. The normal tractions across those planes are called the principal stresses. The state of stress at any point is completely determined by the direction and the magnitude of the principal stresses at the point under consideration. The principal stresses, given in direction and in magnitude, express in the most

[1] General Electric Company, Research Laboratory.

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

general and complete way the elastic state at any given point. The bending moment, the shearing forces, etc., are readily deduced from the direction and the magnitude of the principal stresses. Furthermore, one of the principal stresses always expresses the maximum stress.

2 The notion of principal stress may be illustrated as follows:

3 Consider a spherical element in a solid body. External applied loads will deform this spherical element into an ellipsoidal element (Fig. 1). The axes of this ellipsoid will correspond in direc-

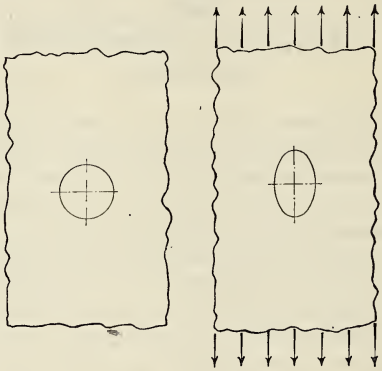

Fig. 1 Ellipsoidal Element Resulting from Subjecting a Spherical Element to Stress

tion and in magnitude to the direction and the magnitude of the principal stresses.

4 The orientation and the form of the ellipsoid, and therefore the direction and the magnitude of the principal stresses, will define the state of stress at the point under consideration.

5 The axes of the ellipsoid represent the largest and the smallest deformation at the point under examination. Correspondingly, the principal stresses give the direction and the magnitude of the maximum and the minimum stress.

6 If the three principal stresses vary from point to point in the structure, the problem to be dealt with is a three-dimensional elastic one. If one of the three principal stresses vanishes throughout, it is a two-dimensional elastic or plane-stress problem.

7 Corresponding to the three- and two-dimensional elastic-stress problems there are also the three- and two-dimensional elas-

tic-strain problems, when the deformations corresponding to the principal stresses are considered.[1]

8 A great number of structural problems (bridge, ship, airplane, plate, dam, etc., construction) are, or their stress analysis may be reduced to, two-dimensional elastic problems.

THE PHOTO-ELASTIC METHOD OF STRESS DETERMINATION

9 As set forth in Par. 1, the state of stress at any point is most completely defined by the direction and the magnitude of the principal stresses. These are, therefore, the elements which we wish to determine for a complete analysis.

10 The photo-elastic method solves the two-dimensional elastic problems. It primarily takes advantage of the double refracting properties shown by isotropic transparent substances when put under stress. The stresses in the structure may therefore be determined from models made of a homogeneous transparent material, and ordinarily on a reduced scale. The stresses in a steel, cement, or any other structure, homogeneous throughout and obeying Hooke's law of linear proportionality between stress and strain, may be readily deduced from the values obtained by the analysis of the corresponding transparent model for the case of two-dimensional elastic problems.

11 If plane polarized light is passed through a stressed specimen of celluloid and afterward through a second nicol prism whose principal section is parallel to the plane of polarization of the original beam of light, only the points where the principal stresses are respectively parallel and perpendicular to the principal sections of the crossed nicols remain dark. This result makes it possible to determine the directions of the principal stresses at any given point. Moreover, this information is needed for the measurements which will be described later.

12 If now circularly polarized light be passed through the specimen, by interference of the two component rays, which in the double-refracting specimen have suffered a relative retardation at each point proportional to the difference in magnitude of the two principal stresses, a colored image is obtained. (Figs. 2, 3, 4 and 5.)

13 By a comparison method, based upon the interposition in

[1] A complete theory of stress and strain may be found in the Treatise on the Mathematical Theory of Elasticity by A. E. H. Love, 3rd ed., chapters i–iv.

the proper direction of a comparison member of constant cross-section, put under uniform tension in a suitable frame (Fig. 6), the value of the difference of the principal stresses at any given point may be read on the dynamometer of the frame.

14 Now, in the two-dimensional elastic problems the transverse deformation, i.e., the deformation along a normal to the plane of the two principal stresses, is proportional to the sum of those two stresses. By means of a lateral extensometer, Fig. 7, we measure this transverse deformation.

15 From the values of the differences and the sums of the principal stresses, the separate values of each of them are computed, thus determining completely the state of stress.

16 A question naturally arising is whether the results obtained on a transparent body such as celluloid hold for structural materials.

17 It is shown by the general discussion of the equations of elastic equilibrium that in the case of strain or plane stress in an isotropic body obeying Hooke's law of linear proportionality between stress and strain, the stress distribution is independent of the moduli of elasticity and consequently of the material of which the body is made. Thus the stress distribution experimentally determined in the case of a celluloid body is the same as it is when the body is made of any other isotropic substance such as iron, steel, etc., obeying Hooke's law, in distribution, direction, and magnitude.[1] Moreover these conclusions derived from the general theory of elasticity have been checked by experiment.[2]

18 The photo-elastic method can be applied to the great majority of structural problems, not only in taking the place of mathematical computation, but particularly in solving those struc-

[1] Except, however, if the body is multiply connected and the resultant applied forces do not vanish separately over each boundary. In this particular case the correction coefficients for passing from one isotropic substance to another may be experimentally determined. (On Stresses in Multiply-Connected Plates, by L. N. G. Filon, British Assn. Report, 1921.)

[2] Photo-Elastic Measurements of the Stress Distribution in Tension Members Used in the Testing of Materials, by E. G. Coker, Excerpt Proc. Inst. C. E. (London), — vol. ccvii, part ii, p. 8.

Photo-Elastic and Strain Measurements of the Effects of Circular Holes on the Distribution of Stress in Tension Members, by E. G. Coker, Trans. Inst. Engrs. & Shipbuilders in Scotland, vol. lxiii, part i, p. 33.

La Photo-Elasticimétrie, ses principes, ses méthodes et ses applications, by Paul Heymans. Bull. Soc. Belge Ing. et Ind., Aug. 1921, pp. 147–154, 165–167, 189–199.

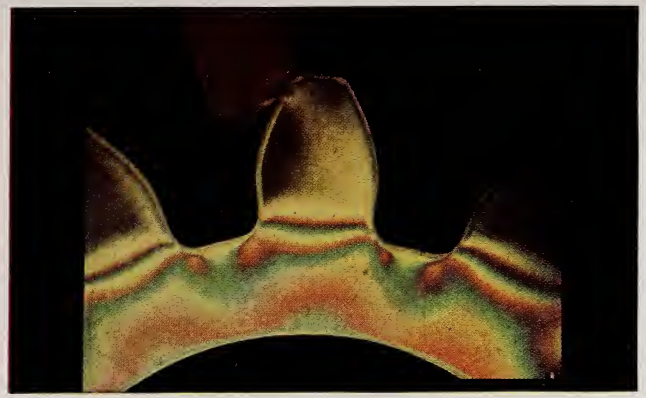

Fig. 2. Colored Image when Normal Inside Pressure Alone is Applied

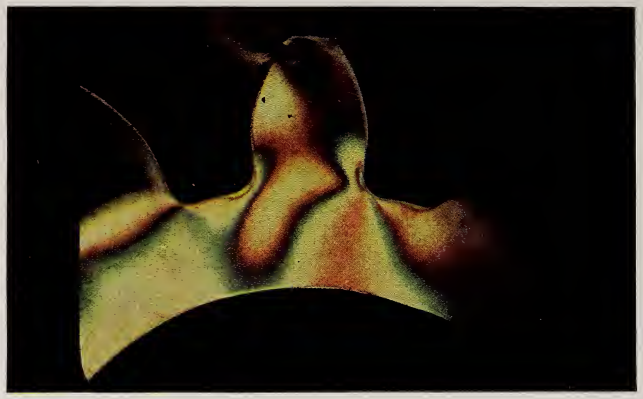

Fig. 5. Colored Image Obtained for Normal Inside Pressure and Reduced Torque

Fig. 3. Colored Image when Both Normal Inside Pressure and Maximum Torque are Applied

Fig. 4. Colored Image when Both Normal Inside Pressure and Maximum Torque are Applied

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tural problems where mathematics becomes too involved to be of help. Moreover it has the great advantage of giving the maximum stress at each point throughout the whole structure, and it there-

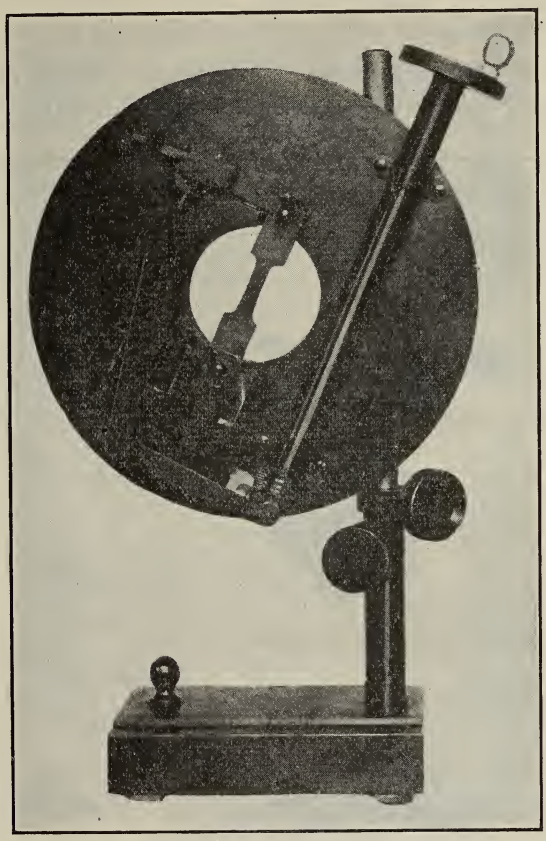


FIG. 6 FRAME FOR COMPARISON MEMBER DESIGNED BY E. G. COKER AND A. L. KIMBALL, JR.

fore offers an effective means of increasing safety and reducing superfluous material.

II — A STUDY OF THE STRESS DISTRIBUTION IN GEAR PINIONS

19 When accidents occur with gear wheels, besides the metallurgical question, three possible causes of failure suggest themselves:

a The gear wheel may not have been properly designed
b It may have failed under an excessive load
c When the pinion was shrunk hot or forced on to a tapered shaft, an excessive inside radial pressure may have been set up.

20 It is easy to see that the ordinary methods of resistance calculations of gear wheels, based on considering the tooth as a

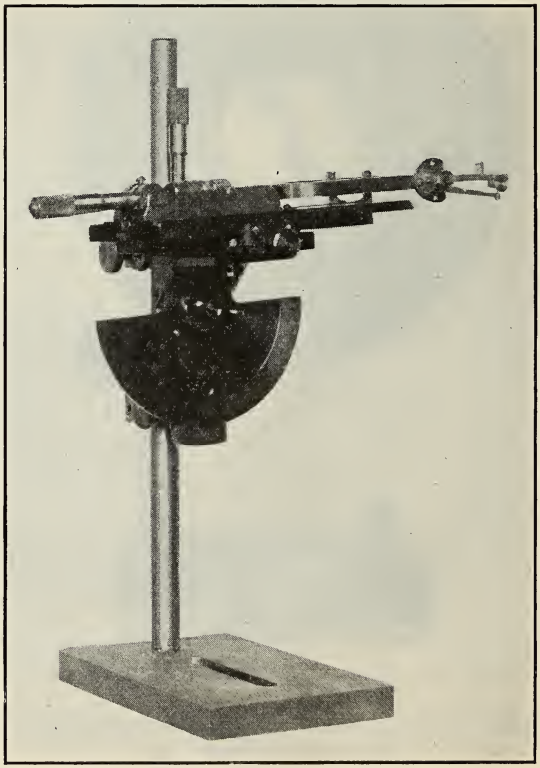


FIG. 7 LATERAL EXTENSOMETER DESIGNED BY P. HEYMANS

cantilever loaded at its end, would not be expected to give reliable and complete information as to stress distribution, not even for the root section of the tooth which is under consideration.

21 Indeed, the shape of the tooth, the curvature at the root, the ratio of the diameter of the pinion bore to the root and outside diameter, the permanent stresses introduced by the placing of the

pinion on the shaft, etc., all affect the stress distribution and the maximum stress. Photo-elastic analysis shows that these factors affect the stresses considerably more than would be expected from present methods of estimating. For standardized pinions the correction coefficients can only partially take account of these factors. For special pinions or for pinions of which more efficient running is required, a photo-elastic analysis seems to be the best if not the only effective way to determine the stress distribution and to locate the maximum stress.

22 A detailed analysis of the stress distribution determined

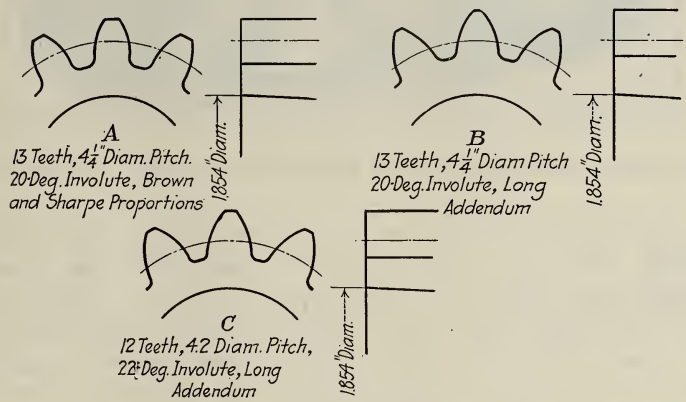


Fig. 8 Tooth Forms of Pinions Subjected to Photo-Elastic Analysis

for different gear pinions and under different loading conditions is given below.

23 The authors wish first to call attention to certain interesting points brought out by photo-elastic analysis, which have been checked by tests carried out on steel sections. These are particularly interesting because they are unexpected.

24 Besides the stress distribution in the different sections of the pinions represented by Fig. 8, the photo-elastic analysis has given as maximum stress under normal inside radial pressure and maximum torque:

80,000 lb. per sq. in. for tooth form *A*
70,350 lb. per sq. in. for tooth form *B*
60,900 lb. per sq. in. for the tooth *C*.

Moreover the 12-tooth pinion shows, besides a smaller maximum stress, a better stress distribution.

25 For steel pinions the maximum stress attained under normal conditions, although high, appears not to be excessive. *Tooth C appeared to be a better design* under normal conditions.

26 The stresses due to shrinking or forcing the pinion on the shaft can only be estimated. The pinion may be assumed to

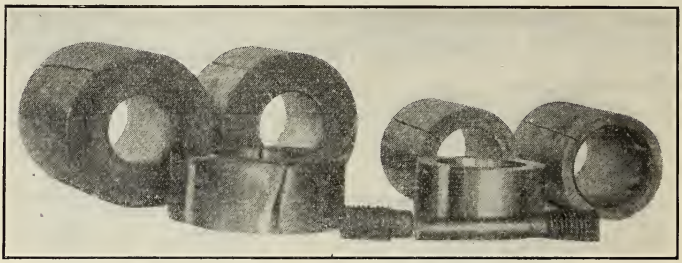


FIG. 9 STEEL RINGS RUPTURED BY BEING FORCED ON TO A TAPERED PLUG

be a plain circular ring, for which case the stresses may be mathematically computed. The stress at any point of the ring as well as the maximum stress in the ring depends upon the lengths of the inside and outside radii. The opinion generally expressed is that for the case of the pinion the maximum stress will be intermediate

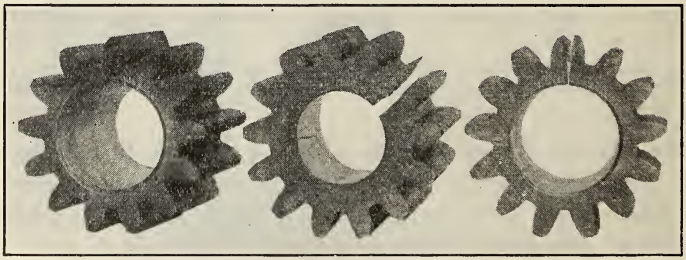


FIG. 10 STEEL PINIONS RUPTURED BY BEING FORCED ON TO A TAPERED PLUG

between the maximum values obtained for rings of which the outside diameters are respectively equal to the root diameter of the tooth and to the outside diameter of the pinion, the inside bore being the same.

27 Photo-elastic analysis shows that *the gear pinion is even weaker than the plain circular ring whose outside diameter is equal to the root diameter of the tooth.* The change of external profile, due to

the presence of the teeth, although requiring an addition of material, weakens the structure.

28 Figs. 9 and 10 show the steel specimens after having been tested by forcing a tapered plug into the bore; and Table 1 gives the rupture load applied to the tapered arbor forced into the bore for the different specimens. These confirm the photo-elastic results.

29 Previous to the photo-elastic investigation of the stresses due to radial inside pressure in pinion sections, fracture due to pure radial inside pressure would have been expected to occur through the minimum radial cross-section.

30 From Fig. 2, representing the color image obtained in the photo-elastic analysis, it appears that *the regions under the teeth*

TABLE 1 RUPTURE LOAD ON ARBOR FORCED INTO SPECIMENS TESTED

	Inside diam., in.	Outside diam., in.	Root diam., in.	Rupture load, lb.
Ring	1.854	3.5	.....	85,000
Ring	1.854	2.5	.....	51,000
Pinion.....	1.854	3.5	2.5	47,000

are under higher stress and that the points at the inside boundary right under the teeth are points of maximum stress.

31 Fig. 10 gives the fractures obtained on steel sections. Two of the sections show fractures right through the thickest layer of material, while all of them started at points where the photo-elastic analysis had revealed maximum stress. The unevenness of the material must account for the deviation of the fracture in one of the cases.

32 Can any statement be made as to the causes of the failure by inspection of the shape of the fracture? In the case in which the authors were interested, the photo-elastic analysis determined the best design. As before said, either the placing of the pinion on the shaft, if carelessly done, for instance by pounding the pinion heavily on the tapered shaft, or excessive torque and blows due to sudden meshing or the taking on of a heavy load, will set up dangerous stresses.

33 The authors' photo-elastic analysis has shown that *the sections of dangerous stresses are different for different values of inside radial pressure and applied torque load.*

34 The fracture shown in Fig. 11 is of an *open* V-shape. Photoelastic analysis shows that *the higher the inside radial pressure becomes,* for a given torque load, *the sharper becomes the V-shape of*

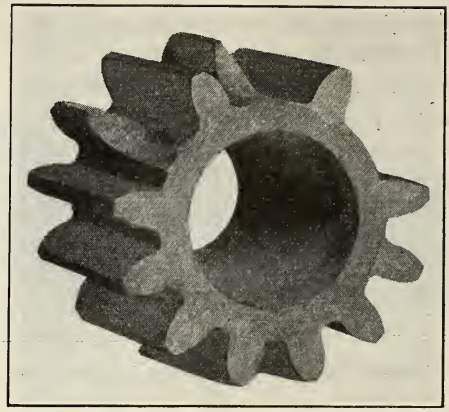


FIG. 11 FATIGUE FAILURES OF TEETH PRODUCED BY EXPERIMENT (WITHOUT RADIAL PRESSURE IN BORE)

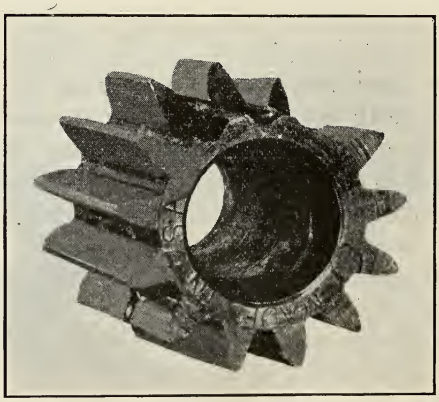


FIG. 12 FATIGUE FAILURES OF TEETH PRODUCED BY EXPERIMENT (WITH HEAVY RADIAL PRESSURE IN BORE)

the section of dangerous stresses. (Fig. 12.) If the fracture is due to too high a torque load, the angle of the V will approach 180 deg. Tests on steel sections have been made with a specially built impact machine.

35 Without inside radial pressure the fracture obtained is a

straight line through the root section of the tooth. With increasing pressures the V-shaped fracture becomes sharper. For an inside radial pressure exceeding the elastic limit, however, the observation does not hold. The reason for this departure from what the photoelastic method had predicted is to be found in the fact that beyond the elastic limit the stress-and-strain relation no longer follows Hooke's law. Therefore the stresses set up in the steel pinions by the shrinking process no longer correspond with those set up in the celluloid model.

36 While the flat shape of the break in Fig. 11 is one limiting case (torque without radial shrinking pressure), Fig. 10 may be considered as the other limiting case (radial shrinking pressure without torque), showing a V-shaped fracture for which the angle of the V has become equal to zero.

37 It may be concluded, then, that the inspection of the fracture may be a means of determining the cause of the failure. In this way, possibly, the responsibility may be established between builder and customer as regards pinion mounting.

THE DETAILED STRESS ANALYSIS

38 *External Forces Applied to the Pinion When in Service.* The pinion is shrunk on to the shaft after having been bored so as to fit the shaft at a temperature of 160 deg. fahr. above normal room temperature.

39 In normal working conditions, the torque load to which the pinion is subjected corresponds to a tractive force of 500 lb. per inch of face of the tooth, tangent in direction to the pitch circle. The whole torque is supposed to be transmitted by a single contact.

40 Calling respectively $\widehat{rr}$ and $\widehat{\theta\theta}$ the radial and the tangential principal stress in a circular ring, of which the outside diameter equals the root diameter of the teeth, the inside bore being the same as the pinion bore, $(\widehat{rr} - \widehat{\theta\theta}) = 28{,}800$ lb. per sq. in. for $\Delta t = 160$ deg. fahr. This value of $(\widehat{rr} - \widehat{\theta\theta})$ is the stress value of the color bands obtained in polarized light (isochromatic bands), and will therefore be used in the stress analysis of the celluloid model to secure the right expansion pressure before the torque is applied. For radial pressures higher than this normal shrinking pressure, the same characteristic of the $(\widehat{rr} - \widehat{\theta\theta})$ value will be used.

41 The tangential tractive force is applied at varying dis-

tances from the root of the tooth, depending upon the point of contact. The most unfavorable conditions arise when this force is applied at the top of a single tooth. Moreover the starting torque load being higher than that realized under normal running conditions, the applied tractive force was brought up from 500 lb. to 1500 lb. per inch of face.

42 Let us for convenience call:

a The *normal* inside pressure, the value of 28,800 lb. per sq. in. for $(\widehat{rr} - \widehat{\theta\theta})$, corresponding to a shrinking pressure due to a temperature variation of 160 deg. fahr.

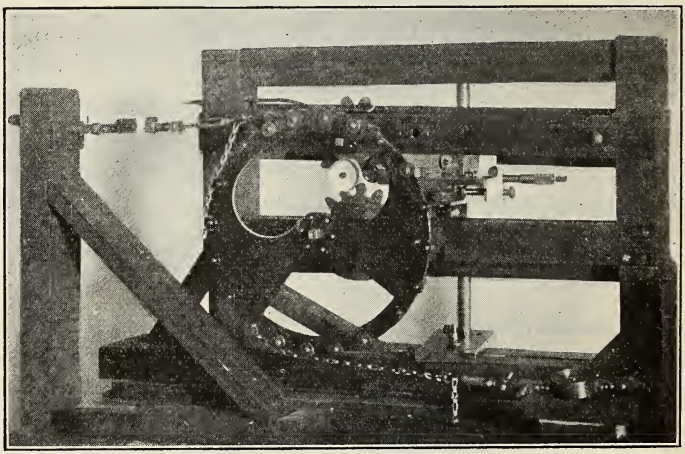


FIG. 13 FRAME USED FOR APPLYING LOADS TO CELLULOID MODELS OF PINIONS

b The *maximum* torque, the torque corresponding to a tractive load F of 1500 lb. applied normally to the contour of the tooth (condition of contact) at the top of one pinion tooth

c The *normal* torque, the torque corresponding to a tractive load F of 500 lb. applied under the same conditions as above

d *Increased* inside pressures, the values of $(\widehat{rr} - \widehat{\theta\theta})$ exceeding the normal inside pressure, as defined above.

43 *The Photo-Elastic Analysis.* Fig. 13 represents the frame used for the loading of the models. A tapered expansion ring is

used to produce the radial inside pressure. The torque is measured by properly mounted dynamometers.

44 The first sets of measurements were made under normal

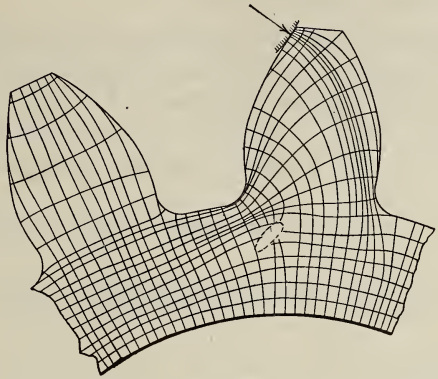


Fig. 14 Lines of Principal Stress Determined by Polarized Light — Normal Inside Pressure and Maximum Torque Load

inside pressure and maximum torque load Fig. 14 represents the lines of principal stress, deduced from the isoclinic bands. The

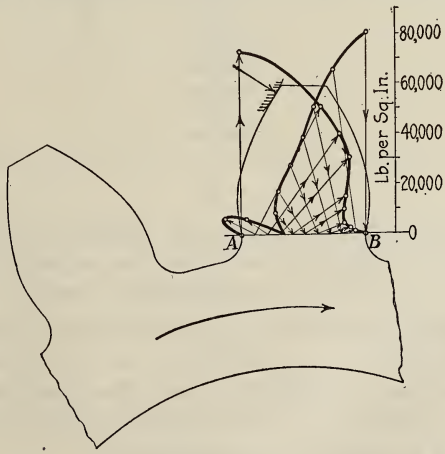


Fig. 15 Curves Showing the Two Principal Stresses in Direction and Magnitude for Points along the Section AB

tangents to these lines represent at each point the directions of the principal stresses.

45 Fig. 2 gives the colored image when the normal inside pressure alone is applied, whereas Figs. 3 and 4 give the image

obtained when both the normal inside pressure and the maximum torque are applied. An optical measurement on the image shown in Fig. 2 allows one to adjust properly the amount of inside pressure before the torque is applied.

46 The determination of the values of the difference $(p - q)$ of the principal stresses is made on the image shown in Fig. 4. One of the two principal stresses vanishes at a boundary where no external forces are applied. In this case the optical measurements of the values of $(p - g)$ give directly the values of the tangential stress.

47 Inside of the body the optical measurements are supplemented by measuring the transverse change of thickness, which gives the values of the sum $(p + q)$ of the principal stresses.

TABLE 2 VALUES OF THE PRINCIPAL STRESSES ACROSS THE MINIMUM CROSS-SECTION OF THE LOADED TOOTH

Tenths of distance AB (Fig. 15) measured from A	p lb. per sq. in.	q lb. per sq. in.
0	0	72,600
0.1	13,850	57,300
0.2	10,450	49,000
0.3	3,710	41,700
0.4	−10,620	25,800
0.5	−20,300	18,700
0.6	−29,000	11,900
0.7	−40,000	9,000
0.8	−51,900	.....
0.9	−65,700	5,320
B	−80,000	0

48 From the values of the principal stresses at a given point it is easy to obtain the stress on a section in any given direction at that point. Moreover, as said before, the two principal stresses represent respectively the maximum and the minimum stress. Thus the larger of the principal stresses will always give at each point the maximum stress in direction and magnitude.

49 At the edges where one of the principal stresses has vanished the values of $(p - q)$ and $(p + q)$ must correspond, i.e., the optical determination of $(p - q)$ and the determination of $(p + q)$ must check.

50 Also as we know the total force acting normally to a given section, the graphical integral of the curve, obtained by plotting

the resultant stresses acting normally to this section, must correspond to the total force. In the case of the pinions the data for such a check are not available, except for the section AB.

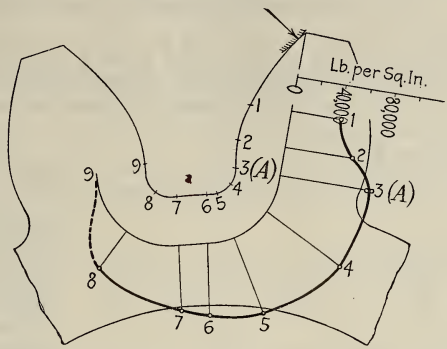


Fig. 16 Tangential Stress at Tension Side — Normal Inside Pressure and Maximum Torque

51 Table 2 gives the values of the principal stresses through the minimum cross-section of the pinion tooth, to which the load is applied. The results given in this table have been plotted in

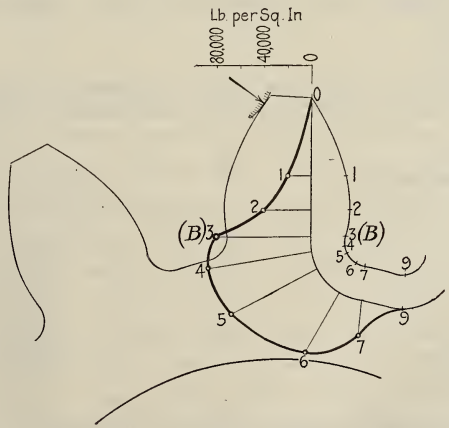


Fig. 17 Tangential Stress at Compression Side — Normal Inside Pressure and Maximum Torque

Fig. 15. At each point where measurements have been made the two principal stresses have been plotted in direction and in magnitude, the arrows serving to distinguish between tension and com-

pression. At the points A and B, $(p - q)$ and $(p + q)$ must check: they differ for A by 0.9 per cent and for B by 0.8 per cent.

52 The maximum tension occurs at A and is equal to 72,600 lb. per sq. in. The maximum compression occurs at B and is equal

TABLE 3 VALUES OF THE TANGENTIAL STRESS AT THE BOUNDARY OF THE LOADED TOOTH — TENSION SIDE

No. of point on Fig. 16	q lb. per sq. in	
1	41,000	
2	54,100	
3(A)	72,300	72,750 [1]
4	73,200	
5	64,800	
6	57,600	
7	54,100	
8	41,000	
9	.....	

[1] Value obtained by taking $\frac{1}{2}[(p + q) + (p - q)]$, the other values being $(p - q)$ measurements.

to 80,000 lb. per sq. in. This difference between the absolute values of these stresses is of course due to the pressure on the inside of the pinion, which affects the tension and the compression stresses differently.

TABLE 4 VALUES OF THE TANGENTIAL STRESS AT THE BOUNDARY OF THE LOADED TOOTH — COMPRESSION SIDE

No. of point on Fig. 17	p lb. per sq. in.	
1	20,500	
2	41,000	
3(B)	79,500	80,000 [1]
4	80,000	
5	82,200	
6	60,000	
7	29,000	
8	0	

[1] Value obtained by taking $\frac{1}{2}[(p + q) + (p - q)]$, the other values being $(p - q)$ measurements.

53 Figs. 16 and 17 give the values of the tangential stresses along the edge of the tooth on which the load is applied. The numerical results of Table 3 have been plotted in Fig. 16, this table giving the tangential stresses at the tension side. Also the numerical

results of Table 4 have been plotted in Fig. 17, this table giving these stresses on the compression side. Since no external load is applied at this side, the optical measurements give the values of the tangential stresses up to the top of the tooth.

54 Table 5 and Fig. 18 give the numerical and plotted values of the stress difference $(\widehat{rr} - \widehat{\theta\theta})$ along the inside boundary of the pinion, the normal inside pressure and the torque load being applied. A circular ring to which a uniform inside pressure is applied will show concentric isochromatic bands. The deflections of those bands (Fig. 2) in the case of the pinion show the disturbance due to the presence of the teeth.

55 When the maximum torque is applied, the values obtained for $(\widehat{rr} - \widehat{\theta\theta})$ give the curve of Fig. 18. The colored images

TABLE 5 VALUES OF $(\widehat{rr} - \widehat{\theta\theta})$ ALONG THE BOUNDARY OF THE BORE

No. of point on Fig. 18	$(\widehat{rr} - \widehat{\theta\theta})$ lb. per sq. in.
1	36,600
2	54,100
3	36,600
4	18,100
5	41,000
6	61,500
7	43,500
8	38,700

as well as the diagrams show that the load applied at the top of one tooth extends its influence as far as the inside boundary of the pinion. The combination of the inside uniform pressure, already disturbed by an irregular outside boundary, with irregularly distributed stresses — tensions in certain parts and compressions in others — due to the torque load, do not of course give a resultant stress distribution which shows any symmetry with respect to the point of contact. The upper pinion being the driving pinion, it may be seen on the colored image (Fig. 3) that the stresses vanish rather rapidly in the withdrawing part, but that the penetration extends much farther into the approaching part.

56 It may also be interesting to point out that there is a zone of zero stress inside of the pinion under the root of the tooth when the torque load is applied. This is often the case at points where lines of principal stresses converge.

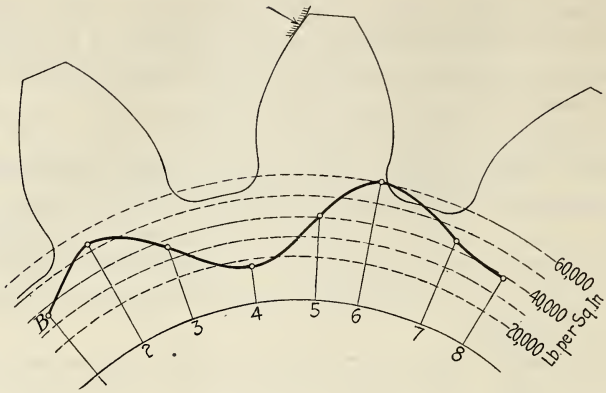


FIG. 18 STRESSES ALONG THE INSIDE OF BORE, WITH NORMAL PRESSURE (28,820 LB.) AND MAXIMUM TORQUE

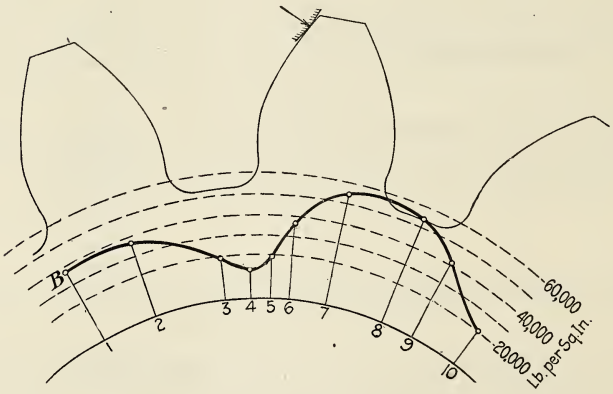


FIG. 19 STRESSES ALONG THE INSIDE OF BORE WITH DECREASED PRESSURE (18,100 LB.) AND MAXIMUM TORQUE

57 The question of engineering interest was to find the relative influence of the factors which affect the maximum stress, and the authors therefore varied the values of:

a The inside normal pressure
b The torque load.

58 The values of $(\widehat{rr} - \widehat{\theta\theta})$ along the inside boundary when the maximum torque load is applied are given in Table 6 and have been plotted in Fig. 19 for the case of reduced inside pressure. The colored image did not show noticeable variation across the mini-

mum cross-section AB and along the outside edges of the main tooth. The influence of the inside pressure within the above-mentioned limit does not affect materially the regions of maximum stress, due in this case to the torque load.

TABLE 6 VALUES OF $(\widehat{rr} - \widehat{\theta\theta})$ ALONG THE BOUNDARY OF THE BORE
(Maximum torque — reduced radial pressure)

No. of point on Fig. 19	$(\widehat{rr} - \widehat{\theta\theta})$ lb. per sq. in.
1	37,600
2	36,600
3	20,500
4	14,550
5	20,500
6	36,600
7	54,100
8	54,100
9	41,000
10	20,500

59 Fig. 5 shows the image obtained for normal pressure and reduced torque. Having applied 0.7 of the maximum torque value, the stresses showed a general reduction in the region of high stress.

TABLE 7 VALUES OF THE TANGENTIAL STRESS ALONG THE BOUNDARY OF THE LOADED TOOTH — TENSION SIDE
(Normal inside pressure — reduced torque)

No. of point on Fig. 20	q lb. per sq. in.
1	39,700
2	51,500
3	58,500
4	57,700
5	56,600
6	51,500
7	38,000
8	19,500

The values of the tangential stresses along the tension side of the boundary of the main tooth are given in Table 7 and are plotted in Fig. 20. This should be compared with the same diagram (Fig. 16) for the case where the full load is applied. The maximum tension has dropped from 73,200 lb. per sq. in. (Table 2) to 57,700 lb. per

sq. in. (Table 7); i.e., it has been reduced to 0.8 of its previous value. The fact that it has only dropped to 0.8 whereas the torque was reduced to 0.7, is explained by the permanent stress due to the inside radial pressure which had been maintained at its previous value. A reduction of the torque load has as a result a reduction of the maximum stress. We shall see later that this is not always the case.

60 When the inside radial pressure is increased in such proportion that without any torque being applied it produces stresses at the outside boundary of the gears of a magnitude approaching

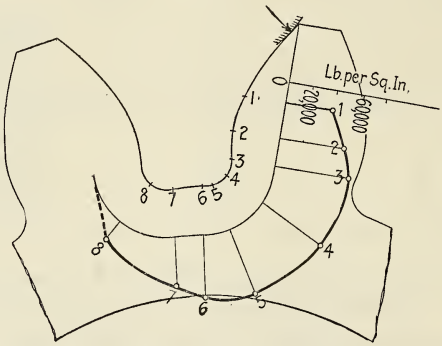


FIG. 20 TANGENTIAL STRESS AT TENSION SIDE — NORMAL INSIDE PRESSURE AND REDUCED TORQUE

that due to the torque load, it will be this internal pressure which will have a preponderant influence.

61 Pinions have been examined with maximum values for $(\widehat{rr} - \widehat{\theta\theta})$ of 60,000 and 81,500 lb. per sq. in. at the inside boundary with the torque load at its normal value of 500 lb. tractive force per inch of face. The tractive force was afterward brought up to its maximum value of 1500 lb.

62 These tests showed that the torque load, when applied to the pinion subjected to those increased radial pressures, affects only the distribution of the stresses. It makes the high stresses extend over a larger area, but it does not increase materially the maximum stress. In these cases the dangerous section is no longer a straight section through the root of the tooth but it follows a V-shaped line, the lower point of which lies toward the inside bore. The sharpness of the angle of the V-shaped fracture at the base of the tooth appears to be due to an excess of radial shrinking pressure.

In practice this excess is due to the pounding of the pinion on to the tapered shaft past its normal position.

63 In this connection a study was made of the stress distri-

Fig. 21 Principal Stresses Across Radial Sections of Tooth — Normal Inside Pressure and Maximum Torque

bution through two radial sections, passing respectively through the points A and B of the minimum cross-section of the main tooth, the points of maximum tension and compression.

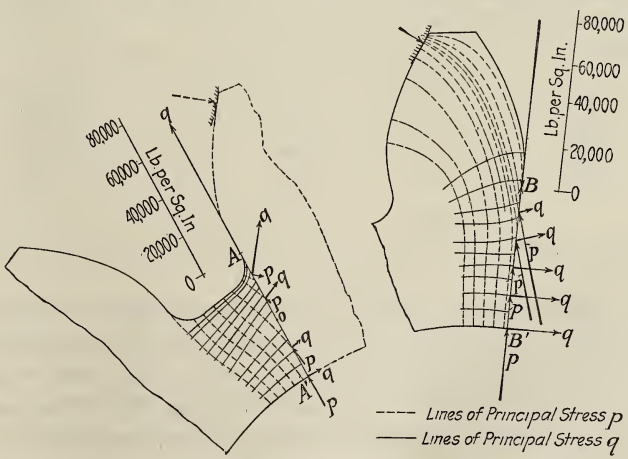


Fig. 22 Principal Stresses in Direction and Magnitude for Same Radial Sections as Those Shown in Fig. 21

64 The values of $(p - q)$ were deduced from the colored image of Fig. 4. Extensometer measurements of $(p + q)$ were made. As before, the scales of both measurements were determined

so that the stresses in the models should represent the stresses in the steel pinion.

65 The maximum torque and the normal inside pressure were applied. Table 8 and Figs. 21 and 22 give the values obtained. Fig. 21 gives the magnitude of the principal stresses along the two sections AA' and BB'. Fig. 22 gives a portion of the lines of principal stress taken from Fig. 14, and for the same sections AA' and BB' shows the two principal stresses plotted in direction and in magnitude.

TABLE 8 VALUES OF THE PRINCIPAL STRESSES ACROSS THE RADIAL SECTIONS PASSING RESPECTIVELY THROUGH THE POINTS A AND B OF THE MINIMUM CROSS-SECTION OF THE LOADED TOOTH

Cross-Section BB', Fig. 21; Distance in inches from point B'	p lb. per sq. in.	q lb. per sq. in.
0.410 (B)	−79,900	0
0.334	−55,800	6,000
0.256	−39,000	15,200
0.179	−32,600	18,400
0.102	−27,100	23,500
Cross-Section AA', Fig. 21; Distance in inches from point A'		
0.410 (A)	0	69,350
0.334	4,350	25,400
0.256	2,700	10,300
0.179	0	0
0.102	−11,350	3,350

66 A good way to visualize the state of stress at a given point is to consider a rectangular element with its sides parallel to the two principal stress directions at that point. By considering such elements along the sections AA' and BB' (Fig. 22) from this viewpoint, one can form a mental picture of how the section is acted upon by the elastic forces.

67 It would require too much space to include in this paper a full discussion and to make a complete report of the results summarized here. The authors trust that the material they have presented will stimulate those interested in this subject to further efforts in the development and use of the photo-elastic method.

68 It seems, finally, almost superfluous to call attention to the comparative ease with which such a stress problem as this can be handled by the photo-elastic method, whereas the use of ordinary engineering methods gives untrustworthy results and the exact mathematical solution based upon the theory of elasticity is impossible.

69 Acknowledgment is due to the Massachusetts Institute of Technology for permission to use in this article certain of the results included in the thesis entitled Photo-Elasticity or Stress Analysis by Means of Optical Methods [1] submitted by Dr. Paul Heymans, University of Ghent, Belgium, as partial fulfillment of the requirements for the degree of Doctor of Science from the Institute.

DISCUSSION

S. Timoshenko. The photo-elastic method used in the experiments described in this paper is by no means new, having found practical application in many technical problems in the last twenty years. It was employed by Mesnager [2] in the stress distribution in elastic arches, by E. G. Coker in the bending of a beam subjected to the action of a concentrated force,[3] in the effects of circular holes on the distribution of stresses, in the stress distribution in tension members used in the testing of materials, in contact pressures, and quite recently in conjunction with K. C. Chakko [4] in the study of the action of cutting tools.

The photo-elastic method of analysis of two dimensional problems of elasticity has a broad application within certain narrow limits and it is important that these limitations be clearly understood in order to use the method correctly and to the best possible advantage. This method of analysis fails where the elastic limit of the material under test is exceeded. In view of this limitation, however, a marked simplification of the work can be made. Because of the proportionality between stress and strain within these limits, the principle of super-position can be used. According to

[1] Abstract in *Tech. Eng'g. News,* May 1922.
[2] *Annales des Ponts et Chaussées,* 1901, 1913.
[3] Edinburgh Roy. Soc. *Trans.,* vol. 41, 1904.
[4] *Proc.,* Institution of Mechanical Engineers, May 1922.

this principle, each force produces its effect independently of any other force and the resultant effect of several simultaneous forces is the vector sum of the individual effects produced by each of them alone. If two observations had been made by the authors — i.e., (1) the stress distribution for some external force acting on the tooth of the pinion and (2) the stress distribution for some pressure acting on the inner diameter — the stress at any point of the pinion, due to any combination of force and pressure (provided that the resultant stress lies within the elastic limit), can be obtained by the above mentioned principle.

It is easy to see how in this manner we have all the information obtainable, instead of just a few isolated facts, and generalizations can be made accordingly. This gives a method of study. It is a simple enough matter to set up the apparatus for getting the stress distribution for a pressure acting on the inner boundary. In the case of the force, however, we have the problem of holding the pinion fixed without materially affecting the stress distribution in which we are actually interested. One way to accomplish this object is to make the pinion and shaft [1] out of one piece of celluloid and to provide the necessary reaction to the force by clamping of the shaft. Another manner of procedure is the following: First, get the stress distribution for the simultaneous action of a given external force and inner pressure, then remove the force so as to obtain the stress distribution due to the inner pressure alone. Evidently, for any cross-section desired, the stress due to the force is the difference between the stresses obtained from experiment (2) and (1).

I am doubtful about the authors' results obtained for stresses along the inner boundary, because of the probability of slight irregularities in the shape of the surface which would produce inequalities of pressure and because of the great probability of initial strains produced by the machining of the sheet of celluloid. It may be well not to attach too much value to the results of these tests, since the presence of a shaft (which necessarily must be longer than the thickness of the pinion) will cause three dimensional stresses at the common boundary of pinion and shaft, for which the photo-elastic method is no longer valid.

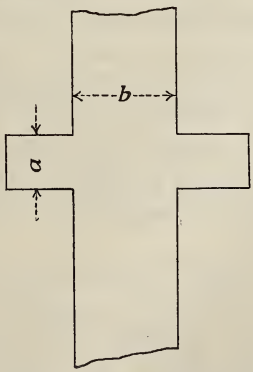
In Par. 58, the authors make the statement that the inside pressure does not materially affect the stresses along the outside

[1] The length of the shaft normal to the pinion will be equal to the thickness of the sheet of celluloid from which the model is cut.

edges of the main tooth. Then, certainly, the stress values for the points 1 and 2 in Fig. 20 should be 0.7 of those in Fig. 16 since the torque values are reduced in the same ratio and since it is the torque only which is responsible for these stresses. But the results of the paper do not confirm this fact. The values for these points, given in Tables 3 and 7, are:

	Table 3	Table 7	Ratio
Point 1	41,000 lb. per sq. in.	39,700 lb. per sq. in.	0.967
Point 2	54,100 lb. per sq. in.	51,500 lb. per sq. in.	0.955

In the study of the effect of pressure acting on the inner diameter, it would have been valuable to compare the pinion with a ring, of which the outer diameter is equal to the diameter at the root of the teeth. We do not expect to find a noticeable difference in the stress on the inner boundaries of these objects, but along the outer boundaries there must be considerable differences due to the presence of the teeth in the case of the pinion. The pinion is comparable with a bar which has sudden changes of cross-section at various points.[1] The conditions at these points, when the bar is put in tension, are similar to those produced at the homologous point in the pinion when pressure is exerted against its inner diameter. We expect the teeth to cause local increases in the stress, but the form of the curve by which they merge into the body of the pinion is of little influence, as in the case of the bar just cited. These local stresses decrease rapidly with increasing distance from the root of the tooth and, as stated in Par. 58, do not affect ma-



[1] We suppose a less than b as in the experiment of E. G. Coker. (See *Proc.*, Mech. Eng. 1921, p. 433.)

terially the stresses across the section which the authors designate as the minimum cross-section. Therefore, I cannot agree with the conclusion of the authors that the difference between the numerical values of the maximum tension (72,600 lb. per sq. in.) and maximum compression (80,000 lb. per sq. in.) is due to the pressure on the inner boundary of the pinion. This difference is more evidently due to the direction of the external force.

The authors, their statement to the contrary notwithstanding, certainly have all the information necessary for checking their work and I do not understand why they neglected to do so. They have, for instance, for the case shown in Fig. 15, the values of the external force and also the values of the stress across the section *AB*, as deducted from their experiments. The resultant force acting on this cross-section can be directly calculated from the known stresses, and it must necessarily balance the external force on the tooth.

In a plastic material such as steel, it is impossible to foretell that a given section will prove weakest at the breaking point of the material, because this section contains the maximum stresses when the body is stressed within the elastic limits. Therefore, the photo-elastic results cannot confirm any rupture tests on specimens. Moreover, no conclusions can be reached from the tests themselves. In all, only three were made and it seems to me that on the basis of these alone, it is just as probable that the breaks through the thicker sections were caused by the initial stresses in the materials as that they were caused by the stresses produced by the applied forces.

The present paper would be of direct benefit to the designing engineer if, for any given tooth shape, it gave him a definite relation between the actual stresses across the " minimum cross-section " of the tooth and those obtained by calculation from the cantilever formula. For example, taking the depth of the tooth above the cross-section considered as equal to $\frac{1}{2}$ in., the thickness equal to $\frac{1}{3}$ in. and the load at the end equal to 1500 lb., the maximum stress obtained from the cantilever formula is

$$p_{max} = \frac{1500 \times \frac{1}{2} \times 6}{(\frac{1}{3})^2} = 40{,}500 \text{ lb. per sq. in.}$$

Comparing this with the stresses 72,600 and 80,000 given by the photo-elastic method (see Table 2 of the paper), we see that the increase of maximum stress due to the local stresses near the root of the tooth, over that given by the cantilever formula, is 79 and 98 per cent.

If the method of study suggested earlier in this discussion had been used, it would have been possible to put the results in such a form that the engineer would know that the maximum stress, produced by a force of P lb. on a given shaped tooth would be equal to KF lb. per sq. in., where F is the maximum stress given by the cantilever formula and K is a constant determined by the researches.

In conclusion, I must point out some errors, which do not, however, affect the principal results of the paper.

In Par. 6, the authors assert that "if one of the three principal stresses vanishes throughout, it is a two-dimensional elastic problem." This conclusion is not correct. If we take, for instance, the torsion of a bar, one of the principal stresses at every point is equal to zero, but the problem is not a two dimensional one.

In Par. 56, it is stated that "several lines of principal stress can only intersect where the principal stresses vanish." This is not true. If we take, for instance, a circular disc subjected to a plane strain symmetrical about the center, the radii of the disc will be lines of principal stress. These radii all intersect at the center of the disc, where the stress is not necessarily zero.

It seems to me that the maximum stresses of 80,000, 70,350, and 60,900 lb. per sq. in. correspond rather to normal radial pressure and *maximum* torque, than to normal radial pressure and *normal* torque, as it is stated in Par. 24.

J. O. Madison.[1] We have on the road today three different motors; in the subway, 200-hp. motors; on the elevated, approximately 125-hp. motors; and on the surface, motors ranging from 4 to 65 hp.

In large motors there is plenty of material between the bottom of tooth and the shaft, from an inch and up to an inch and thirteen-sixteenths; and there is very little trouble holding the pinions on and very little trouble with splitting of pinions or even the breaking of teeth. They are boiled in water and put on hot with one blow of a 10-pound sledge.

On the elevated we have smaller pinions, and the distance is a trifle under an inch from the bottom of the tooth to the shaft. These pinions are secured by boiling and by hammering them on with three blows of a 10-pound sledge, and we are finding con-

[1] Engineer Car Equipment, Interborough Rapid Transit Co., New York, N. Y.

siderable trouble with them; we are having quite a number of tooth breakages and occasionally pinions split through the hub material. On the surface car pinions, where there is the smallest amount of material under the tooth, the pinions are put on by heating and also by hammering them on with approximately 8 to 10 blows of the 10-pound sledge. With these there is very great breakage, but we cannot apparently get along without putting them on thus securely, because if we do not hammer them on they loosen up in service.

It would appear, therefore, that we need some very strong material with good elongation on motor pinions, particularly where there is a small amount of material under the tooth.

ALPHONSE A. ADLER. We should know all the fundamental facts about the design of pinions. The models should be rotated, as in practice, and a moving picture obtained in order to be sure that maximum stress occurs only at the time the teeth get into action. Then we would be in a position to determine what assumptions must be made in the elastic theory in order to obtain a useful formula.

E. O. WATERS. It seems to the writer that there would be quite a field for this photo-elastic method in connection with the relation between stresses and pressure distribution and wear of material. Take, for example, the case of gear and pinion contact. The interesting question is whether that method would show the same stress distribution for new gears as for gears that have been in service for some time. Taken in the case of a bearing supporting a shaft, if the bearing is new, the contact might be at a few points, whereas if it was a well-worn bearing, the pressure ought to be fairly uniformly distributed. In those cases is it possible to use this method with actual samples of the material in question, using very thin sections? It seems that the experiment would be more direct if the same material was used as in actual service.

PAUL HEYMANS, in answer to many questions, brought out the following points regarding this paper:

The celluloid models were mounted on steel shafts.

No scale for the values of the colors expressed in stress magni-

tude can be given independently of the nature of the material nor of the thickness at the point under consideration.

Besides the nature of the material, the interference effects, covering the colored image, depend upon the thickness of the material traversed by the ray. For models of constant thickness, the same table of correspondence between interference effect and stress holds. If the model is of varying thickness, a table of correspondence in function of the thickness has to be used for the interpretation of the colored images. However, in the photo-elastic analysis such a scale would not generally be of satisfactory accuracy. A comparison member put under uniform tension has ordinarily to be used.

The light band running along the edge of certain of the colored pictures is due to a superficial shrinkage caused by evaporation at the surface of certain of the volatile components of the celluloid after the model has been machined. This " edge effect " does not usually cause much inconvenience. Mr. T. H. Frost of our laboratory has recently gone into the developing of methods to prevent this shrinkage, and has obtained satisfactory results for celluloid of known composition.

No key-ways have been put in the models. The models represented a section of the pinion at the larger end of the tapered bore. The key-way originates at the opposite end and dies off gradually to nothing at the end which was analyzed.

The photo-elastic method allows the investigation of the stress distribution in all models through which it is possible to pass the ray of polarized light in a direction parallel to one of the three principal stresses. This is, of course, always the case for a two-dimensional elastic stress problem. As has been observed, the greatest majority of problems of engineering interest are two dimensional elastic stress problems.

The stress analysis carried out for these pinions has been made on homogeneous material, and consequently holds for homogeneous steel. However, as has been pointed out, case hardening does not change the values of the elastic constants entering in the general stress-strain relations, and consequently does not alter the isotropy of the material on elastic point of view. Therefore, the stress analysis in isotropic celluloid holds.

The photo-elastic stress analysis is facilitated when the model presents two faces parallel to the plane of the principal stresses which have to be determined. This is, however, not indispensable.

We may mention that we have in progress the stress analysis of the crankshaft of an airplane engine which by its shape and the numerous lubricating conduits represents a piece of very irregular contour.

Mr. Waters' remarks can be answered in a general way by observing that the photo-elastic method implies the building of a model in transparent material, in which the stresses may be studied under any given loading conditions. Regarding the specific question of the relation between stresses and wear of material in general and in the case of gears in particular, there is certainly quite a field for the photo-elastic method. The wear of material in gears and shafts, altering the meshing conditions and the conditions of the bearings, as affecting the stress distribution, could be investigated simply by considering a certain number of specific cases. It is, so far, not possible to investigate directly the stresses in steel pinions. The method is based on the effect of the stressed regions on polarized light passing through the specimen. This implies an investigation on transparent models. Regarding the use of very thin steel sections, it must be observed that the interference effects, originating the colored images, increase, up to a certain limit, with the thickness. Extremely thin sections would, therefore, at least disturb considerably the accuracy of the photo-elastic measurements.

In fact, numerous fields of application of the photo-elastic method to very fundamental structural problems are already absorbing our attention, however anxious we may be to consider more of the problems which are proposed to us. We are gradually extending our laboratory with the hope that we will be able to respond to more of the questions which are of a troublesome nature to engineers.

The Authors. S. Timoshenko states that the method fails when the elastic limit of the material under test is exceeded. Of course this is true. The whole of the classical theory of elasticity applies only to cases of stress distribution where Hooke's Law is obeyed. Par. 17 of our paper covers this point.

It is, of course, true that the same stress distribution can be obtained by combining the stress distribution due to radial pressure and that due to the torque load according to the law of superposition. The stress distribution due to a torque load alone

would be difficult to determine, as stated. Probably the best way to obtain this is to subtract the stress distribution due to the radial pressure alone from that due to the combined radial pressure and torque load. We then have the basis for combining these stress systems in any way desired.

In regard to the presence of the shift disturbing the plane stress distribution so that photo-elastic method fails, we would say that it may disturb slightly, but the photo-elastic method itself is only accurate to within two or three per cent and the effect mentioned is, in all probability, much less than this and has therefore been ignored.

In answer to the suggestion that we compare the pinion with a ring, the outer diameter of which is equal to the diameter at the root of the tooth, we refer to Par. 26 and 27 of our paper where that comparison is described. The authors disagree with the statement "it is just as probable that the breaks through the thicker section were caused by the initial stresses, etc.," in reference to rupturing steel pinions by forcing tapered plugs in them. The facts found were that three tests showed failures each time starting directly under the pinion which is just the point shown by the photo-elastic method to have the greatest principal stress difference. It is, to say the least, a striking result. We agree with Dr. Timoshenko to the extent that further experiment is desirable.

The definition in Par. 6 is only a definition and correct as it stands, and not a "conclusion" as stated by Dr. Timoshenko. In confirmation of this we refer him to Love's Mathematical Theory of Elasticity, Third Edition, Par. 94.

It is true that where the principal stresses intersect the stress is not necessarily zero.

No. 1860

THE ELASTICITY OF PIPE BENDS

By Sabin Crocker, Detroit, Mich.
Associate-Member of the Society
and
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Junior Member of the Society

In this paper the authors develop in detail mathematical formulas connecting the deflection of different types of pipe bends with the force producing deflection, and also expressing the stress set up as a function of the observed deflection and the constants of the bend. They describe experiments made to check force vs. *displacement, and present charts for use in selecting the proper size and shape of bends to take care of a given expansion, and to determine the force exerted by a bend when in a given state of deflection. In general they recommend bends of larger dimensions than are ordinarily used, and their figures are said to lie on the side of safety. In some cases the bends tested showed a greater flexibility than anticipated, due to minute folds formed on the compression side when bending pipe to short radius.*

THE use of expansion loops, offsets in the line, right-angled turns and similar devices to furnish flexibility for expansion and contraction in pipe lines resulting from temperature changes, has been general practice for years. Such bends, which utilize the elasticity of the pipe itself, are very commonly used for high-pressure work in preference to slip joints, corrugated expansion joints, or swivel offsets in which screwed elbows are arranged to turn on pipe threads. The use of pipe bends reduces internal friction in the piping by providing easy turns, eliminates unnecessary fittings and joints, and facilitates clearing other pipes and structural interferences. The required flexibility can be obtained in some cases by introducing in the line a member consisting of some form of expansion bend like those illustrated in Figs. 4, 5 and 6, or, what is more common, it may be accomplished by directional changes in the whole line of piping. An ideal condition for the latter exists where two long runs of pipe can be placed at right angles to each other, as is

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frequently done, for example, in a main steam line from boiler to turbine.

2 Up to the present time the disposal of expansion and contraction in power-plant piping has been largely a matter of judgment rather than of rational calculation on the part of the designer. This has been due to a general lack of knowledge or interest regarding the principles and methods involved in calculating the forces and stresses resulting from the expansion of piping. However, if certain rather fundamental principles of engineering mechanics are brought into play, it is usually possible to determine numerical values with fair accuracy during the progress of design, and make use of this knowledge to improve the piping layout. As the drift toward higher steam pressures and temperatures in turbine practice continues, the problem becomes more and more important. Since there is also a tendency for the steam turbine to grow smaller relative to the size of the steam line to which it is connected, there is a danger that excessive stresses will be set up in the turbine casing as a result of expansion in the pipe line unless the piping designer knows how to calculate the forces involved. This paper is presented to set forth a method of attacking the problem. It also shows wherein the authors' methods and test results differ from the work of previous contributors to the subject and the extent to which they check earlier work.

3 The amount of expansion to be cared for in a pipe line can be accurately computed, provided the temperature range between extreme hot and cold conditions, the length of pipe, and the coefficient of expansion for the material are known. Having computed the amount of expansion or elongation of the pipe, the problem resolves itself into determining whether the piping arrangement under consideration has sufficient "spring" to absorb this elongation without producing either undue fiber stress in the material or excessive forces tending to tear the piping loose from its anchors. At the same time the proportions of the line must be economical and be adjusted to clear any structural interferences in the plant. Wall thickness and diameter of pipe, radius of curvature, and general layout all affect the flexibility which can be obtained with a given run of pipe.

4 The inadequacy of the usual methods of steam-pipe design has been brought forcibly to the attention of The Detroit Edison Company by several minor mishaps during the past few years. Fortunately, none of these was of a serious nature, but they did serve to show that piping laid out according to the best modern practice may be subject to forces of unexpected proportions. Thus in one instance

unexpectedly large reactions caused a 14-in. high-pressure steam main to tear loose from what was supposed to be an adequate anchorage. This incident, and others of a somewhat similar nature, led those responsible for design and operation to ask themselves whether such failures could not be avoided by more intelligent design. Consequently, in undertaking the design of the company's Marysville power plant, where the elongation per hundred feet of steam pipe amounts to almost seven inches and the steam pressure is 300 lb. per sq. in., it was decided that pipe expansion must be cared for in a rational manner if such a thing were possible. Accordingly, the investigation here reported was undertaken to determine the fundamental relations between deflection, force resisting deflection, and stress set up in expansion pipe bends of various shapes. These relations were first determined mathematically, and later checked by physically testing bends of shapes which readily lent themselves to measurement of the quantities involved. The results thus obtained were then applied to the more complicated conditions in actual pipe lines. As a result it was possible to design the piping layout for the first section of the Marysville power plant, which has just been put in operation, so that the following results are obtained without allowance for cold springing:

Maximum fiber stress in pipe wall.....6900 lb. per sq. in.
Maximum thrust against pipe anchorage 680 lb.
Maximum thrust against turbine flange 580 lb.

If credit is taken for cold springing, the maximum stress and forces given above are further reduced. These values have been obtained without making the expansion bends unnecessarily large.

PREVIOUS WORK BY OTHERS

5 Before proceeding along independent lines an extensive search was made for existing literature, which disclosed that very little had been written on the subject. However, the following articles are of special interest:

(*A*) Formänderung und Beanspruchung federnder Ausgleichröhren (Strains and Stresses in Expansion Bends), by Prof. A. Bantlin, *Zeitschrift des Vereines deutscher Ingenieure.* Jan. 8, 1910, p. 43.

(*B*) Expansion of Pipes, by Ralph C. Taggart, Trans. Am. Soc. C. E., paper No. 1167, Dec. 1910.

(*C*) Elasticity and Endurance of Steam Pipes, by C. E. Stromeyer, *Engineering*, June 19, 1914, p. 857 (from a paper read before the Institution of Naval Architects).

(*D*) Pipe Bends, Their Growing Use and Efficacy, *The Valve World*, Oct., 1915 (published by Crane Co.).

(*E*) The Design of Pipe Bends for Expansion in Pipe Lines, by J. G. Stewart, *Power*, May 10, 1921, p. 742.

6 Reference (*A*) deals with the comparison of physical tests and calculated results for bends similar to our double-offset expansion U-bends. The results are tabulated for comparison and explanation of discrepancies rather than for use by a designer laying out pipe lines.

7 Reference (*B*) treats principally of the use of straight pipe and fittings in making up expansion loops. The results are in general similar to those of the present authors for straight lengths of pipe.

8 Reference (*C*) describes a series of failures of steam pipes in service, due to repeated strains beyond the elastic limit. A set of formulas similar to those presented in this paper was worked out to show what elongations could be cared for without eventually producing failure.

9 Reference (*D*) is a report of physical tests on pipe bends made by the Mechanical Experts Department of Crane Company, and published in their organization paper, *The Valve World.* A series of curves deduced from their experimental data and giving the amount of expansion required to produce a fiber stress of 15,000 lb. per sq. in. in different-shaped bends was included for the use of those interested in pipe designs.

10 Reference (*E*) gives a method of computing fiber stress similar to that of Reference (*C*), and a chart for reading the elongation corresponding to 15,000 lb. per sq. in. stress for different bends.

11 It is worth comment that while the formulas which follow were worked out independently of Reference (*C*) and before Reference (*E*) was published, all three sets are practically identical. The comparison of theoretical formulas with actual test results, and the rather complete design graphs worked out in this paper, take the subject a step beyond the point reached by earlier publications. The fact that none of the previous investigators has published all of the mathematical steps leading up to his final formulas, will make the derivations given by the authors of special interest to anyone having occasion to work with the formulas.

12 Table 1 is a summary of such data published in the above references as were in shape to reduce to a common form for comparison with the work of the present authors. It will be noted that the formulas of References (*C*) and (*E*) agree very closely with those of

this paper, while Reference (*D*), which has been widely quoted during the past seven years, differs considerably from the others except for the double-offset expansion U-bend. The following extract is quoted from Reference (*D*):

> A 180-deg. or U-bend has twice the expansive value of a 90-deg. or quarter-bend of the same size and radius, and an expansion U-bend, four times the expansive value of a quarter-bend or twice that of a U-bend. A double-offset expansion U-bend has five times the expansive value of a quarter-bend, two and one-half times that of a U-bend, and one and one-fourth times that of an expansion U-bend.

In other words, the relative ability of these particular bends to take up expansion without exceeding a safe stress is stated by Reference

TABLE 1 SUMMARY OF DATA ON PIPE BENDS

	C = Stress Index C, in $S = C\,\frac{D\,\Delta\,E}{R^2}$				K = Flexibility Index K, in $\Delta = K\,\frac{FR^3}{EI}$			
Reference	Quarter bend	U-bend	Expansion U-bend	Double-offset expansion U-bend	Quarter bend	U-bend	Expansion U-bend	Double offset expansion U-bend
(*C*) C. E. Stromeyer[1] . .	1.404	0.318	0.106	0.0495	0.356	1.571	9.426	34.54
(*D*) Crane Co.[2]	0.2	0.1	0.05	0.04	—	—	—	—
(*E*) J. G. Stewart[1]	1.404	0.318	0.106	0.045	0.356	1.571	9.42	37.9
Crocker & Sanford, Calculated	1.404	0.318	0.106	0.0427	0.356	1.571	9.42	39.88
Crocker & Sanford, Test.	—	—	0.048 to 0.106	0.042 to 0.0427	—	—	9.42 to 20.8	39.9 to 40.7

[1] For references, see Par. 5 of paper.
[2] Formulas deduced from curves published in *The Valve World*, Oct. 1915.
For shape of bends see Fig. 12, or Figs. 1, 4, 5 and 6.

Definitions of Symbols

C = stress index (numerical values given in left half of table). (Formulas 3, 7, 9, 11.)
K = flexibility index (numerical values given in right half of table). (Formulas 2, 6, 8, 10.)
S = maximum bending stress in pipe wall, lb. per sq. in.
D = outside diameter of pipe, in.
R = mean radius to which bend is formed, in.
E = modulus of elasticity (30,000,000 lb. per sq. in. for steel).
I = moment of inertia of section, taken normal to center line of pipe of which bend is formed, in.^{4}.
F = force causing deflection of bend, lb.
Δ = deflection of flange of bend = elongation of pipe line due to change in temperature, in.
= total expansion to be absorbed by bend, i.e., total expansion calculated for a straight run of pipe with length equal to distance between anchors which occur nearest to bend under consideration.

(*D*) to be in the ratio of 1 : 2 : 4 : 5, while the calculated ratio given by References (*C*) and (*E*), and checked by the present authors, is 1 : 4.4 : 13.2 : 32.9, assuming that the quarter-bend is connected to the pipe line so that the thrust of the line acts along the axis of the pipe in the bend at the joint, as shown in Fig. 1.

13 Referring to the stress index of Table 1, Reference (*D*) agrees quite closely with the others' results for double-offset expansion U-bends, but gives much higher values for the expansion cared for by quarter-bends, U-bends, and expansion U-bends than do the others. It will also be noted that the stress and flexibility indexes for the double-offset expansion U-bend given by Reference (*E*) differ slightly from those given by the present authors. This is because the two bends considered are not of exactly the same shape.

14 A peculiarity of bends noted in Reference (*A*) was also observed in some of the bends tested by the authors, i.e., that the elasticity found by test exceeded the computed elasticity. Reference (*A*) describes tests on three kinds of bends formed to a shape which resembles a double-offset expansion U-bend, the first variety being bent from solid rod stock, the second from steel pipe, while the third was a hollow iron casting similar to a pipe bend. Tests of the first and third varieties checked very closely with calculated results, while the bend fabricated from a straight length of pipe was found to be much more elastic than the calculations indicated. The explanation given in Reference (*A*) is as follows:

> We therefore establish:
>
> (1) That for the *solid* lyre-shaped bends a satisfactory conformity of calculated and measured value can be demonstrated.
>
> (2) That, on the contrary, for the lyre-shaped *pipe* the actual deflection was very much greater than the calculated value.
>
> In addition to the reasons cited, which result in a difference between measured and calculated results the slight effect of which was demonstrated in the experiments with solid bends, there must exist a further reason, peculiar to the conditions which obtain in the pipe, to which the very great difference in measured and calculated value can be attributed.
>
> It can be deduced from the sketch of the test piece (a double-offset expansion U-bend), that it has deep waves or folds on the inner sides of the bends, that is, on the sides subjected to compression in the manufacture of the bend, these folds originating during the bending of the particular pipe length from which the bend is manufactured.
>
> It is clear that the pipe bends which have these waves must possess much less resistance to the compressive forces which they have to withstand than would a smooth pipe bend without such waves. The waves or folds, by reason of their great number, will operate in an elastic manner despite their infinitely small depth; this must result in a very much greater total deflection of the bend

than indicated by calculation, which of course assumes a pipe with smooth walls. If this view is correct, the phenomenon of the great elastic deflection of the expansion bends is in perfect accord with the purpose for which such an expansion bend is placed in long steam lines, and the greater elasticity yielded by the above-mentioned waves should be regarded as a valuable property possessed by the expansion bend.

In order to determine by experiment the correctness of the assumption regarding the effect of the waves — to prove it by calculation would necessitate very complicated and involved equations of little value — a cast-iron lyre-shaped pipe was made in which the waves or folds on the compressed side of the bend were absent. The pipe, of about 200 mm. (7.87 in.) inside diameter and about 18 mm. (0.7 in.) wall thickness, had the same bend radius as the steel pipe. It was furnished by Sulzer Bros., Winterthur, and represented an outstanding example of foundry technique when we consider the difficulty presented by the production of a test bend of this complicated design and of such dimensions. The deflections of this cast-iron pipe were determined by the same method and by the same equations as were employed in the case of the steel pipe.

From these experiments with the cast-iron pipes, together with those conducted with the solid steel lyre-shaped bodies, we are led to conclude that the large deflections of the steel pipes must be attributed chiefly to waves or folds upon the compressed side of the pipe.

15 This conclusion is supported by the results presented here, all of the test bends formed to a radius of five pipe diameters or less having a flexibility in excess of calculated values, while, with one exception, those formed to a radius of six diameters or more gave results agreeing closely with those obtained by the formulas. This would seem to indicate that the excessive elasticity was due to puckers or folds in the pipe wall produced in the process of bending, the tendency being to thin the pipe wall at the outer circumference of the bend and form minute crinkles or waves at the inner circumference. These conditions are more pronounced the shorter the radius of curvature, and in the case of the bend described in Reference (*A*), which was bent to a radius of only three to four pipe diameters, the waves were plainly visible. In American practice, where the minimum allowable radius of curvature is limited to five or six pipe diameters, these waves are not so visibly evident, nevertheless we may assume that they are present and have the effect of increasing the flexibility of bends formed to too short a radius.

MATHEMATICAL ANALYSIS

16 A pipe bend may be considered as a beam of special form, one end of which is fixed and the other end of which is free to move in the direction of a force acting upon it. Since the bend is an elastic

body, the movement of its free end will be proportional to the force acting, within the elastic limit of the material.

17 Consider any pipe bend, as in Fig. 1, showing a quarter-bend, with force F acting as shown. Flange A is fixed; flange B is free to move in the direction in which the force is acting. Let Δ be the total deflection of flange B measured in the direction of F, and caused by the force as it changes from 0 to F; then, considering the problem on an energy basis, the total work done by F will be $\frac{1}{2}F\Delta$,

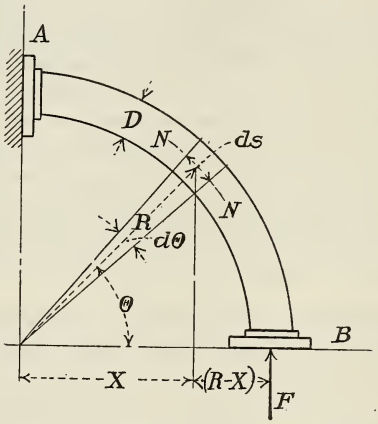


Fig. 1 Quarter-Bend, Force Axial

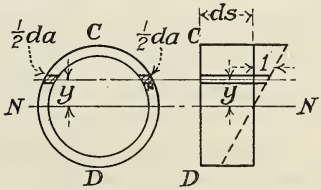


Fig. 2 Elements of Pipe Bend

or the average force times the distance moved. The work done by F is also equal to the internal work of distortion in the bend. To obtain an expression for the work of distortion, consider an element of the bend of length ds (Figs. 1 and 2). Let M = bending moment at this section. The unit bending stress S existing at a distance y from the neutral axis N-N is My/I, I being the moment of inertia of the section. The total stress ΣS on a fiber of cross-section da located at this distance from the neutral axis is $Myda/I$. Let l be the elongation of this fiber and E the modulus of elasticity; then —

$$E = \frac{\text{unit stress}}{\text{unit strain}} = \frac{\frac{My}{I}}{\frac{l}{ds}}, \text{ whence } l = \frac{Myds}{EI}$$

The work of distortion is —

$$\frac{1}{2} l\Sigma S = \frac{1}{2} \frac{Myds}{EI} \frac{Myda}{I} = \frac{M^2ds}{2EI^2} y^2da$$

Integrating across the section, we have, for the work on the element ds, $W = \int_C^D \frac{M^2ds}{2EI^2} y^2da$, and since $\int_C^D y^2da = I$ by definition, it follows that $W = \frac{M^2ds}{2EI}$. The total work of distortion for the entire bend then is $\int_B^A \frac{M^2ds}{2EI}$, and since this is equal to the work done by F, we have, $\frac{1}{2} F\Delta = \int_B^A \frac{M^2ds}{2EI}$, from which —

$$\Delta = \frac{\int_B^A M^2ds}{FEI} \quad \ldots\ldots\ldots\ldots \quad [1]$$

This expression is a general one and will apply to any type of bend. In the work which follows it has been developed for several of the most common types.

THE QUARTER-BEND, FORCE AXIAL AT ONE END

18 Consider Fig. 1, with a force F acting as shown.

$$\Delta = \int_B^A \frac{M^2ds}{FEI}$$

Let R be the radius to which the bend is formed. Then for the element ds, $M = F(R - X)$, and

$$\Delta = \int_B^A \frac{F^2(R - X)^2ds}{FEI} = \frac{F}{EI} \int_B^A (R - X)^2ds$$

Let θ be the variable angle between the X-axis and the radial line to the element ds, and let $d\theta$ be an increment of that angle. Then $X = R \cos \theta$ and $ds = Rd\theta$. Substituting,

$$\Delta = \frac{F}{EI} \int_0^{\frac{\pi}{2}} (R - R \cos \theta)^2 Rd\,\theta = \frac{FR^3}{EI} \left(\frac{3\pi}{4} - 2\right)$$

$$= 0.3562 \frac{FR^3}{EI} \quad \ldots\ldots\ldots\ldots \quad [2]$$

19 The maximum bending stress existing in the pipe wall of the bend for a given deflection may be derived from this expression. As before, the unit stress existing at a distance y from the neutral axis of the pipe is My/I. Let D = outside diameter of the pipe, then since the greatest stress in any section occurs at the outermost fiber, the unit stress S at this point equals $MD/2I$. For the quarter bend the maximum moment occurs at A, where $M = FR$, so the maximum stress $S = FRD/2I$. From the deflection formula [2] for this case $F = \dfrac{\Delta EI}{0.3562R^3}$. Substituting,

$$S = 1.404\,\frac{D\Delta E}{R^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

It is obvious that for a given value of Δ the maximum fiber stress is proportional to the outside diameter of the pipe from which the bend

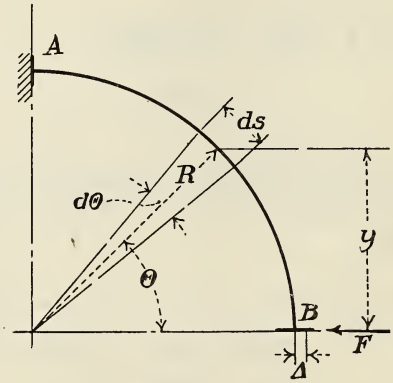


Fig. 3 Quarter-Bend, Force Perpendicular

is made and that it is independent of the thickness of the pipe wall, i.e., the "weight" of the pipe. This will be found true for all cases considered.

THE QUARTER-BEND, FORCE PERPENDICULAR

20 Consider also the quarter-bend with a force F acting as shown in Fig. 3. Flange A is fixed, Δ is the deflection of flange B measured in the direction of F, $\Delta = \dfrac{\int_B^A M^2 ds}{FEI}$, $M = Fy$, $y = R\sin\theta$, and $ds = Rd\theta$. Substituting,

$$\Delta = \frac{FR^3}{EI}\int_0^{\frac{\pi}{2}} \sin^2\theta d\theta = \frac{\pi}{4}\frac{FR^3}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

For a given deflection the maximum unit bending stress S in the quarter-bend occurs at A. $S = MD/2I = FRD/2I$, and from the deflection formula [4] $F = 4\Delta EI/\pi R^3$. Substituting,

$$S = \frac{2D\Delta E}{\pi R^2} = \frac{0.636D\Delta E}{R^2} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

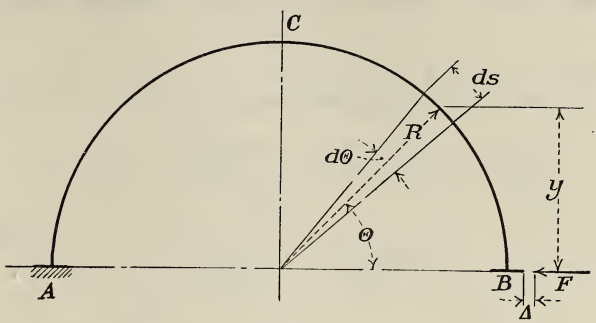


FIG. 4 PLAIN U-BEND

THE U-BEND

21 Consider Fig. 4, with a force F acting along the line AB as shown. Flange A is fixed. Δ is the deflection of flange B measured in the direction of F as before.

$$\Delta = \frac{\int_B^A M^2 ds}{FEI}, \quad M = Fy$$

and—

$$\Delta = \frac{\int_B^A F^2 y^2 ds}{FEI} = \frac{F}{EI}\int_B^A y^2 ds$$

also— $\qquad y = R \sin\theta \quad \text{and } ds = Rd\theta$

Substituting,

$$\Delta = \frac{FR^3}{EI}\int_0^\pi \sin^2\theta d\theta = \frac{\pi}{2}\frac{FR^3}{EI} = 1.571\frac{FR^3}{EI} \quad . \; . \; . \; . \quad [6]$$

For a given deflection the maximum bending stress in the U-bend occurs at C. Let D be the outside diameter of the pipe.

$$S = \frac{MD}{2I} = \frac{FRD}{2I}, \text{ and } F = \frac{2\Delta EI}{\pi R^3}$$

Substituting,

$$S = \frac{D\Delta E}{\pi R^2} = \frac{0.318 D\Delta E}{R^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

THE EXPANSION U-BEND

22 This bend is more complicated than the ones already considered and must be treated in a slightly different manner. The

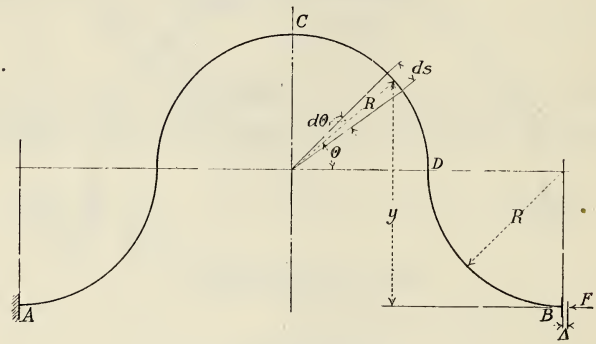


FIG. 5 EXPANSION U-BEND

force F is acting along the line AB as shown in Fig. 5. Flange A is fixed. Δ is the total deflection of flange B measured in the direction of F. This total deflection Δ may be considered as the sum of the deflections due to bending in the various portions of the bend. Let Δ_1 be the deflection of flange B due to bending in the portion DB and Δ_2 its deflection due to bending in the portion CD. Then since the two halves of the pipe are identical, $\Delta = 2(\Delta_1 + \Delta_2)$.

23 Consider the portion DB and assume D as fixed. Then, from the quarter-bend analysis, Formula [2],

$$\Delta_1 = \frac{FR^3}{EI}\left(\frac{3\pi}{4} - 2\right)$$

Now consider the portion CD and assume it fixed at C. Δ_2 is the deflection of B due to bending in CD only, and is measured in the direction of F.

$$M = Fy, \quad \Delta_2 = \frac{\int_D^C M^2 ds}{FEI} = \frac{F}{EI}\int_D^C y^2 ds$$

$$ds = Rd\theta; \quad y = R + R\sin\theta = R(1 + \sin\theta)$$

Substituting,

$$\Delta_2 = \frac{F}{EI}\int_0^{\frac{\pi}{2}} R^2(1 + \sin\theta)^2 R d\theta = \frac{FR^3}{EI}\left(\frac{3\pi}{4} + 2\right)$$

$$\Delta = 2(\Delta_1 + \Delta_2) = 2\frac{FR^3}{EI}\left(\frac{3\pi}{4} - 2 + \frac{3\pi}{4} + 2\right)$$

$$= \frac{9.42\,FR^3}{EI} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [8]$$

24 The maximum unit bending stress S in the expansion U-bend occurs at C. Let D be the outside diameter of the pipe,

$$S = \frac{MD}{2I} = \frac{F(2R)D}{2I} = \frac{FRD}{I} \text{ and } F = \frac{\Delta EI}{3\pi R^3}$$

Substituting,

$$S = \frac{D\Delta E}{3\pi R^2} = \frac{0.106 D\Delta E}{R^2} \quad \ldots\ldots\ldots \quad [9]$$

THE DOUBLE-OFFSET EXPANSION U-BEND

25 The analysis for this bend is similar to that for the expansion U-bend. Force F acts as shown in Fig. 6, along the line AB. Flange A is fixed. Δ is the total deflection of flange B measured in the direction of F. Let Δ_1 = deflection of flange B due to bending in the portion DB and Δ_2 its deflection due to bending in the portion CD. Then $\Delta = 2(\Delta_1 + \Delta_2)$ as before.

26 Consider the portion DB and assume D as fixed.

$$M = Fy; \quad \Delta_1 = \frac{\int_B^D M^2 ds}{FEI} = \frac{F}{EI}\int_B^D y^2 ds$$

$$y = R - R\cos\theta = R(1 - \cos\theta); \quad ds = Rd\theta$$

Substituting,

$$\Delta_1 = \frac{F}{EI}\int_0^{\frac{3}{4}\pi} R^2(1 - \cos\theta)^2 R d\theta$$

$$= \frac{FR^3}{EI}\left(\frac{9\pi}{8} - \sqrt{2} - \frac{1}{4}\right)$$

$$= \frac{1.873 FR^3}{EI}$$

27 Now consider the portion CD, and assume it fixed at C. Δ_2 is the deflection of B due to bending in CD only, and is measured in the direction of F.

$$M = Fy \quad \Delta_2 = \frac{\int_D^C M^2 ds}{FEI} = \frac{F}{EI}\int_D^C y^2 ds$$

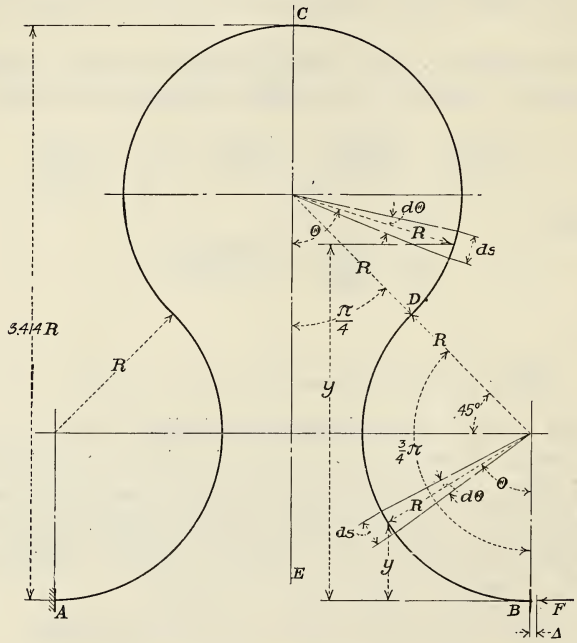


Fig. 6 Double-Offset Expansion U-Bend

$$y = R + 2R \sin 45° - \cos\theta \quad (\theta \text{ measured from the axis } CE)$$
$$= R\,(1 + \sqrt{2} - \cos\theta)$$
$$ds = Rd\theta$$

Substituting,

$$\Delta_2 = \frac{F}{EI}\int_{\frac{\pi}{4}}^{\pi} R^2\,(1 + \sqrt{2} - \cos\theta)^2\, Rd\theta$$
$$= \frac{FR^3}{EI}\left(\frac{21\pi}{8} + \frac{3\pi}{\sqrt{2}} + \sqrt{2} + \frac{7}{4}\right)$$

$$= \frac{18.065FR^3}{EI}$$

$$\Delta = 2(\Delta_1 + \Delta_2)$$

$$= 2\left(\frac{1.873FR^3}{EI} + \frac{18.065\,FR^3}{EI}\right)$$

$$= \frac{39.876FR^3}{EI} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [10]$$

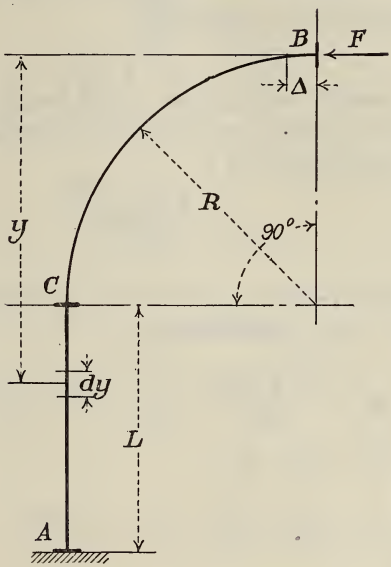


Fig. 7 Quarter-Bend and Tangent, Force Axial

28 The maximum bending stress in the double-offset expansion U-bend occurs at C. Let D = the outside diameter of the pipe and

$$S = \frac{MD}{2I} = \frac{F(2 + \sqrt{2})\,RD}{2I}$$

From the deflection formula [10], $F = \dfrac{\Delta\, EI}{39.876R^3}$. Substituting,

$$S = 0.0427\,\frac{D\Delta E}{R^2} \quad \ldots\ldots\ldots\ldots \quad [11]$$

QUARTER-BEND AND TANGENT, FORCE AXIAL

29 Consider Fig. 7, with force F acting as shown. Flange A is fixed. Let Δ be the total deflection of flange B, measured in the direction of F. Let Δ_1 be the deflection of flange B due to bending in the portion CB, and Δ_2 the deflection due to bending in the portion AC. Then $\Delta = \Delta_1 + \Delta_2$.

30 If we consider C as fixed, then from the quarter-bend analysis formula [2], $\Delta_1 = 0.3562 \dfrac{FR^3}{EI}$.

31 Now consider the portion AC of length L and assume it fixed at A. Δ_2 is the deflection of B due to bending in AC only, and is measured in the direction of F.

$$\Delta_2 = \frac{\int_C^A M^2 ds}{FEI}$$

The element dy is at a distance y from the line of action of the force F, $M = Fy$, and $ds = dy$. Substituting,

$$\Delta_2 = \frac{\int_R^{R+L} F^2 y^2 dy}{FEI} = \frac{F}{EI} \times \frac{(R+L)^3 - R^3}{3}$$

$$\Delta = \Delta_1 + \Delta_2$$

$$\Delta = \frac{F}{3EI}\left[(R+L)^3 + 0.069R^3\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

32 The maximum fiber stress in the pipe wall occurs at A.

$$S = \frac{MD}{2I}, \quad M = F(L+R)$$

Substituting,

$$S = \frac{1.5D\Delta E(L+R)}{(R+L)^3 + 0.069R^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [13]$$

QUARTER-BEND AND TANGENT, FORCE PERPENDICULAR

33 Consider Fig. 8 with force F acting as shown, parallel to AC. Flange A is fixed. Let Δ be the total deflection of flange B, measured in the direction of F. Let Δ_1 be the deflection of the flange B due to bending in the portion CB and Δ_2 the deflection due to bending in the portion AC, then $\Delta = \Delta_1 + \Delta_2$.

34 If we consider C as fixed, then from Formula [4], for the quarter-bend acted upon by a perpendicular force, $\Delta_1 = \frac{\pi}{4}\frac{FR^3}{EI}$

35 Now consider the portion AC of length L and assume it fixed at A. Δ_2 is the deflection of B due to bending in AC only, and is measured in the direction of F.

$$\Delta_2 = \frac{\int_C^A M^2 ds}{FEI}, \text{ and } M = FR$$

Substituting,

$$\Delta_2 = \frac{FR^2}{EI}\int_0^L ds = \frac{FR^2L}{EI}$$

$$\Delta = \Delta_1 + \Delta_2$$

$$= \frac{FR^2}{EI}\left(\frac{\pi R}{4} + L\right) \quad \ldots\ldots\ldots\ldots \quad [14]$$

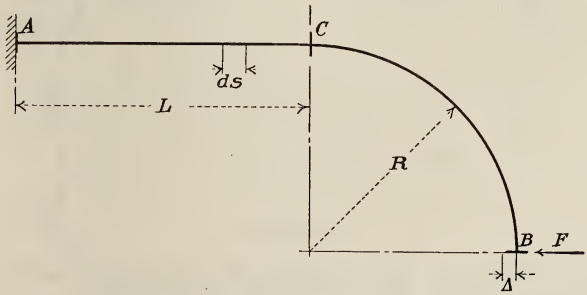


Fig. 8 Quarter-Bend and Tangent, Force Perpendicular

36 The maximum fiber stress in the pipe will occur at A. $S = \frac{MD}{2I}$ and $M = FR$. Substituting,

$$S = \frac{FRD}{2I} = \frac{D\Delta E}{2R\left(\frac{\pi R}{4} + L\right)} \quad \ldots\ldots \quad [15]$$

QUARTER-BEND WITH PERPENDICULAR FORCE ON ADJACENT TANGENT

37 Lastly, consider Fig. 9. Force F acts as shown, parallel to AC. Flange A is fixed. Δ is the total deflection of flange B, measured

in the direction of F. Let Δ_1 be the deflection of flange B due to bending in the portion BD, Δ_2 the deflection due to bending in the portion DC and Δ_3 the deflection due to bending in the portion CA. Then $\Delta = \Delta_1 + \Delta_2 + \Delta_3$.

38 If we consider D as fixed and BD to be of length L_1, then from the deflection formula for a cantilever beam,

$$\Delta_1 = \frac{FL^3_1}{3EI}$$

39 Now consider the portion DC and consider it fixed at C. Δ_2 is the deflection of B due to bending in DC only and is measured

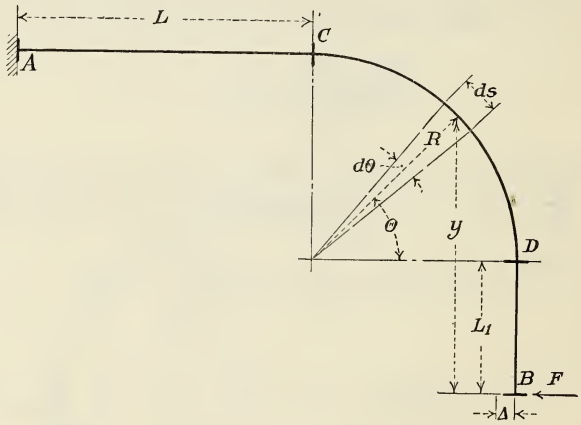


FIG. 9 QUARTER-BEND AND TANGENTS

in the direction of F. Let θ be the variable angle between the X-axis and the radial line to the element ds, and let $d\theta$ be an increment of that angle. y is the distance from ds to the line of action of F.

$$\Delta_2 = \frac{\int_D^C M^2 ds}{FEI}$$

$$ds = Rd\theta \text{ and } y = L_1 + R \sin\theta$$

Substituting,

$$\Delta_2 = \frac{F}{EI}\int_0^{\frac{\pi}{2}} (L_1 + R \sin \theta)^2 Rd\theta$$

$$= \frac{FR}{EI}\left(\frac{\pi L_1^2}{2} + \frac{\pi R^2}{4} + 2L_1R\right)$$

40 Δ_3 is the deflection of B due to bending in the portion AC of length L.

$$\Delta_3 = \frac{FL(L_1 + R)^2}{EI}$$

$$\Delta = \Delta_1 + \Delta_2 + \Delta_3$$

$$\Delta = \frac{F}{EI}\left[L(L_1 + R)^2 + R\left(\frac{\pi L^2{}_1}{2} + \frac{\pi R^2}{4} + 2L_1R\right) + \frac{L_1{}^3}{3}\right] \quad . \quad . \quad . \quad [16]$$

The maximum fiber stress occurs at A.

$$S = \frac{F(R + L_1)D}{2I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where F is determined from [16].

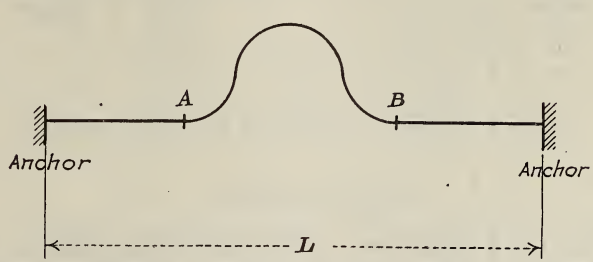


Fig. 10 Actual Piping Layout

GENERAL

41 In like manner the formula for any shape of bend that the designer may use can be derived from the fundamental equation $\Delta = \frac{\int M^2 ds}{FEI}$ by considering the bend a section at a time. The method of attack should be clear from the foregoing.

42 In each case which has been taken up, flange A has been considered as fixed. Such a condition would exist if the line to which A is connected were anchored at that point, or if A were a flange at the turbine. As a matter of fact, the entire analysis is really based on the movement of flange B with respect to A, and it makes no difference whether A is actually fixed or is permitted to move. F is the net force with which the pipe line pushes against the flange B when the line is hot. Such cases as pictured in Figs. 3, 4, 8 and 9, where the pipe line is represented as pushing at right angles to the pipe in the bend, are encountered when the fitting at B is a tee, a cross or

a short-radius ell. For design purposes the total expansion to be cared for between anchorages is determined from known or assumed temperatures, length of pipe, and coefficient of linear expansion. The length of pipe used is a nominal length equal to the actual linear distance between anchorages. In effect, it is as though a line of pipe such as that shown in Fig. 10 were assumed to be like that in Fig. 11, with the total movement due to the expansion of a straight pipe of length L applied at the flange B.

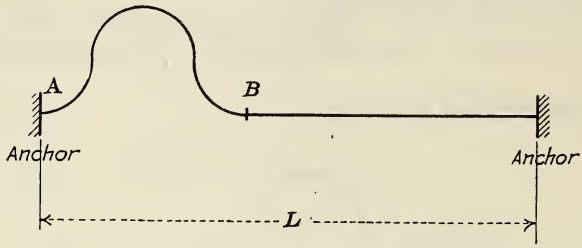


Fig. 11 Layout as Assumed for Calculation

DESIGN GRAPHS

43 For the convenience of the power-plant designer, the formulas for several of the standard bends are represented graphically in Figs. 12, 13 and 14. The bends considered are the quarter-bend, U-bend, expansion U-bend, and double-offset expansion U-bend with the forces applied as shown in the small figures on the graph and corresponding to Figs. 1, 4, 5 and 6. The forces and stresses set up in bends made up of pipe sizes up to and including 20 in., and bent to radii up to and including 120 in., may be read from these graphs without the necessity of solving the formulas.

44 Fig. 12 gives the maximum bending stress in the pipe wall of the bend for any expansion in the pipe line which must be cared for. Inasmuch as the stress is proportional to the outside diameter of the pipe for any given value of Δ, and is independent of its weight, this diagram serves for all weights of pipe. This diagram is drawn up for 8-in. pipe and values for that size may be read directly. For instance, an 8-in. expansion U-bend, formed to a radius of 60 in. and required to take up an expansion of 0.7 in. in the pipe line, would have a maximum bending stress set up in it of 5380 lb. per sq. in.

45 A sample problem will indicate the method to be used for other sizes of pipe. Since the stress is proportional to the diameter

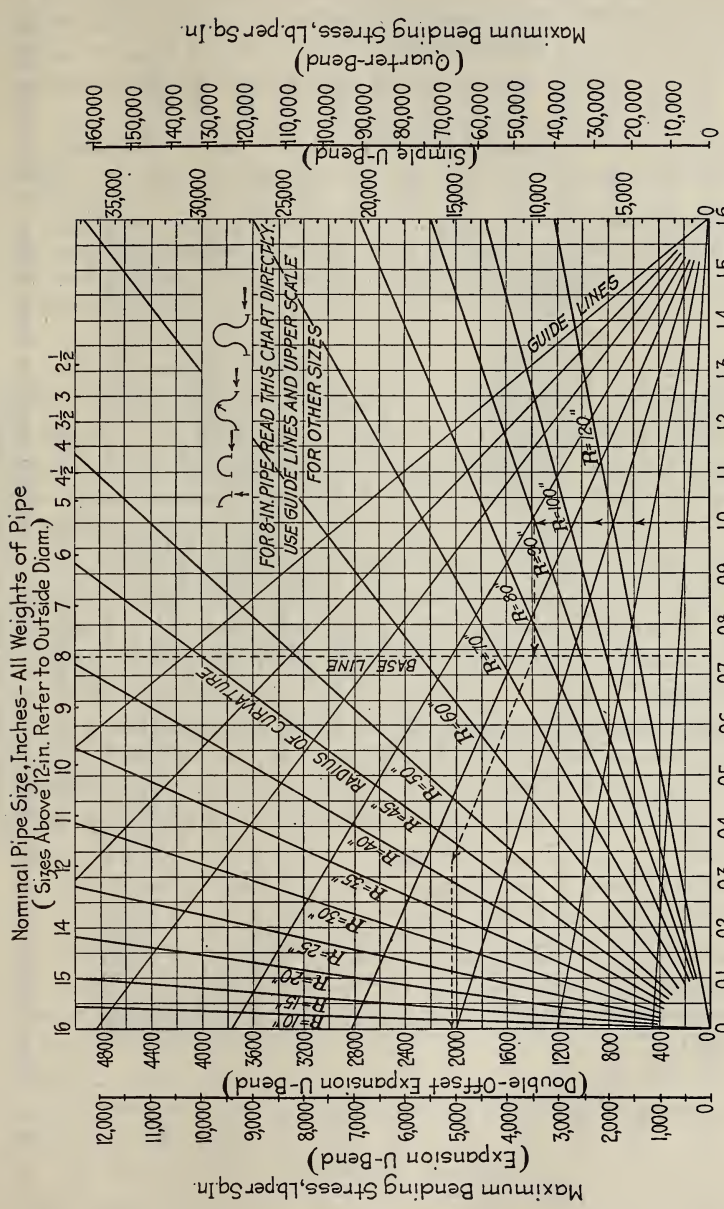


FIG. 12 STRESS VS. DISPLACEMENT FOR VARIOUS SHAPES AND SIZES OF PIPE BENDS

(Graphical Solution of Formulas [3], [7], [9], and [11].)

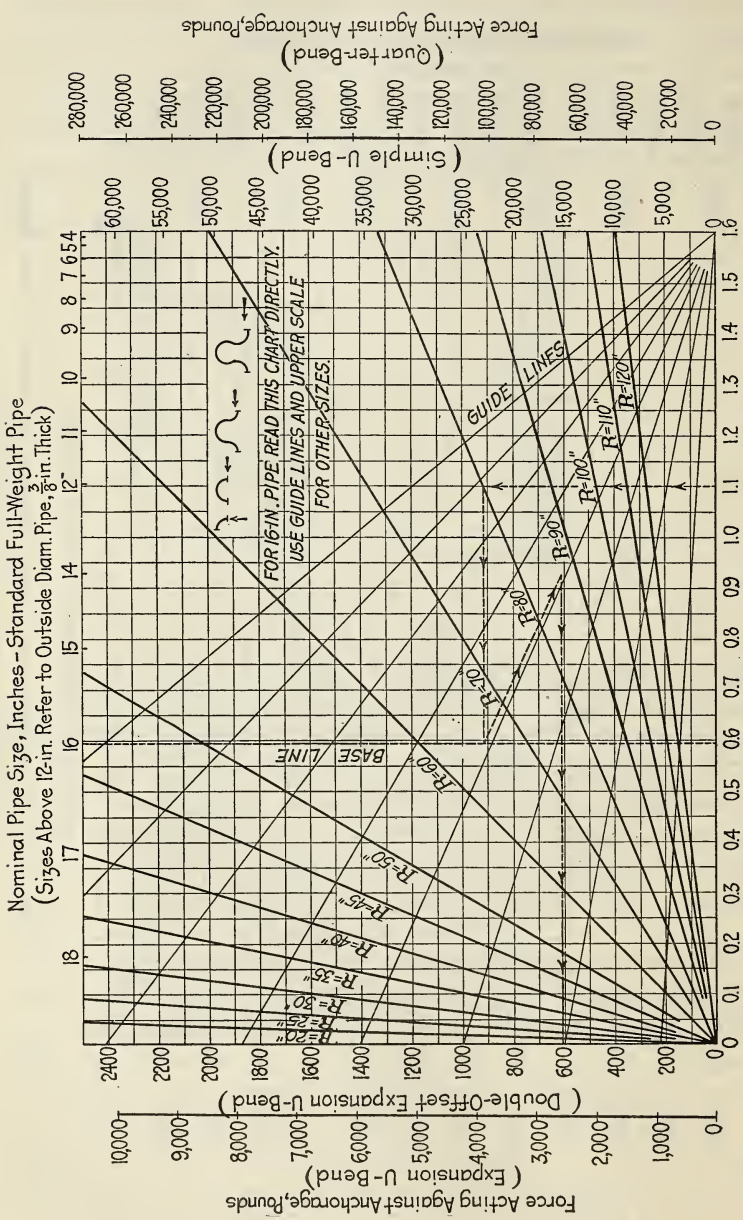


FIG. 13 DISPLACEMENT VS. FORCE EXERTED ON VARIOUS SHAPES AND SIZES OF PIPE BENDS — STANDARD PIPE (Graphical Solution of Formulas [2], [6], [8], and [10].)

of the pipe, the value of the stress for any size of pipe may be obtained after finding the stress in an 8-in. pipe. The proportion between stresses can be made directly on the diagram by following the radial lines diverging from its lower right corner.

SAMPLE PROBLEM: A 12-in. double-offset expansion U-bend having a radius of 90 in. is to take up an expansion of 1 in. Required, the maximum bending stress in the bend.

From the intersection of $\Delta = 1$ in. and $R = 90$ in., extend horizontally to the vertical line under 8-in. pipe, which is referred to as the "base line," and then obliquely (with a line passing through lower right-hand corner) until under 12-in. pipe on the upper scale. The result is 2040 lb. per sq. in.

46 Since the stress is directly proportional to the displacement, this diagram may be used for values of Δ which fall beyond its range by multiplying both stress and displacement scales by the same factor. For instance, if a 5-in. expansion is to be considered, the stress for an expansion of 1 in. should be read on the graph and multiplied by 5.

47 Figs. 13 and 14 give the forces set up in the pipe line when pipe bends are used to take up the expansion. As these relations are functions of the moments of inertia of the pipe sections, the internal diameters affect the results, and Fig. 13 has been drawn for "standard weight" and Fig. 14 for "extra heavy" pipe. These diagrams are for 16-in. pipe and values for this size may be read directly. The scale of pipe sizes at the top of diagram is laid off according to the moment of inertia of the pipe section, as the force to produce a given deflection in the bend is proportional not to the diameter, but to the moment of inertia of the pipe.

48 After finding the force existing for a given deflection in the 16-in. pipe, the value for any other size of pipe may be obtained by using the radial lines diverging from the lower right corner, and the vertical base line under 16-in. pipe, for making the proportion. The solution for a sample problem is indicated by the dash lines in Fig. 13. A double-offset expansion U-bend made of 14-in. O.D. pipe, $\frac{3}{8}$ in. thick, formed to a radius of 80 in., is required to take up an expansion of 1.1 in. in the pipe line. From the diagram, the force acting against the pipe anchorage is 610 lb.

49 Since the force is directly proportional to the displacement, these diagrams also may be used for values which fall beyond their range by considering both force and displacement scales as multiplied by the same factor. It should be noted that in using these

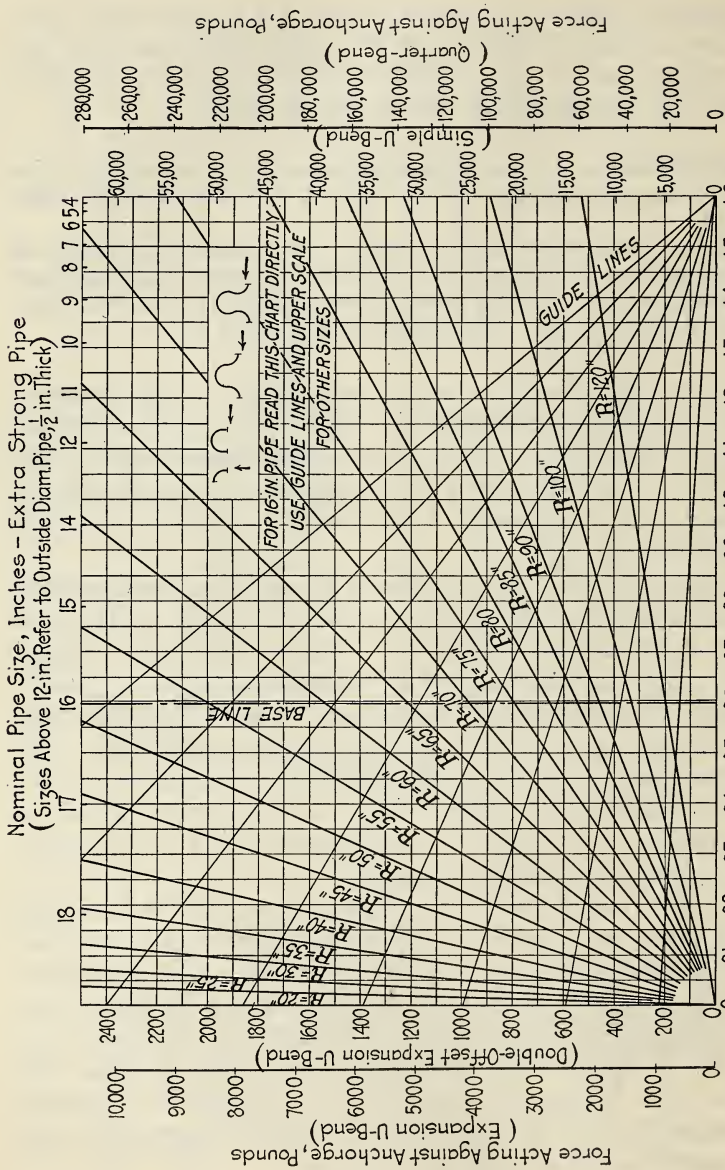


FIG. 14 DISPLACEMENT VS. FORCE EXERTED ON VARIOUS SHAPES AND SIZES OF PIPE BENDS

(Graphical Solution of Formulas [2], [6], [8], and [10].)

diagrams the pipe size refers to the size of pipe used in the bend and has nothing to do with the size of pipe in the rest of the line.

METHOD OF TEST

50 The method used in determining experimentally the relation between force and deflection is shown in Fig. 15, which is a photograph of the test set-up for a 6-in. double-offset expansion U-bend. The left end of the bend was held rigidly in place, while the right end was deflected by means of a hydraulic jack. The amount of deflection was measured with a steel tape, and the force causing deflection was computed from the readings of a pressure gage attached to the reservoir beneath the ram of the jack. The far end of the bend was held in place by securely bolting it to the

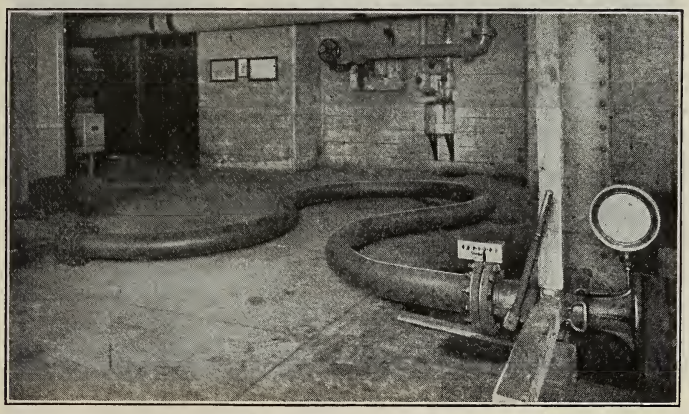


Fig. 15 Set-Up for Test of Elasticity of 6-in. Double-Offset Expansion U-Bend

flange of a straight length of pipe, the end of this pipe resting against a concrete foundation and being prevented from moving sideways by timber braces. A blank flange was bolted to the free end of the bend, this flange having a small screwed fitting at its center, which fitted into the head of the ram and prevented it from slipping. The base of the jack rested against concrete steps and the barrel of the jack was braced with timbers. The flange at the free end of the jack was supported on rollers and the top of the loop was supported in a similar way so that it was free to move as force was applied. The jack used had a capacity of 30 tons, the ram being $3\frac{1}{2}$ in. in diameter and having a maximum lift of 9 in.

51 The pressure required to overcome initial friction before any movement of the bend took place, was determined roughly by lightly wedging the point of a pencil between the movable head and the stationary barrel of the jack, so that it was supported horizontally, and then carefully pumping the jack and noting the pressure at which the pencil was released, a very slight movement of the head being sufficient to let the pencil fall. Before the gage pressure was recorded the end of the bend was jolted up and down so as to relieve any binding action that might be present.

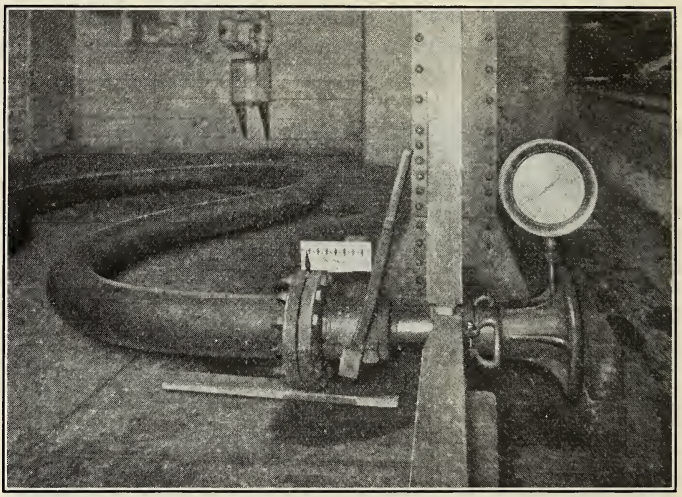


Fig. 16 Test Set-up with Ram Extended Against the Expansion Bend

52 In measuring the deflection, one end of the steel tape was hooked over the head of a small stove bolt screwed into the top of the pipe in the bend near the fixed flange. Measurements were taken to a small chisel mark on the free end of the bend near the flange. In this way errors in reading the deflection due to a movement of the end supports were eliminated. The distance between the ends of the bend was measured first with no force applied to the free end, and then with the head of the jack in different positions according to the pressure applied, noting the gage pressure on the jack each time. Before each reading was taken the jack was lined up so that the line of action of the force passed through the center of the fixed flange. The free end of the bend was also jolted up and

down before each reading, to relieve any binding action. Care was taken not to use forces which would stress the material at the top of the loop beyond its elastic limit. The ram was pushed out to almost the elastic limit of the pipe material and released several times for each bend, as many intermediate readings as possible being taken each time.

53 The only bends tested were expansion U-bends and double-offset expansion U-bends, because these shapes were easy to sup-

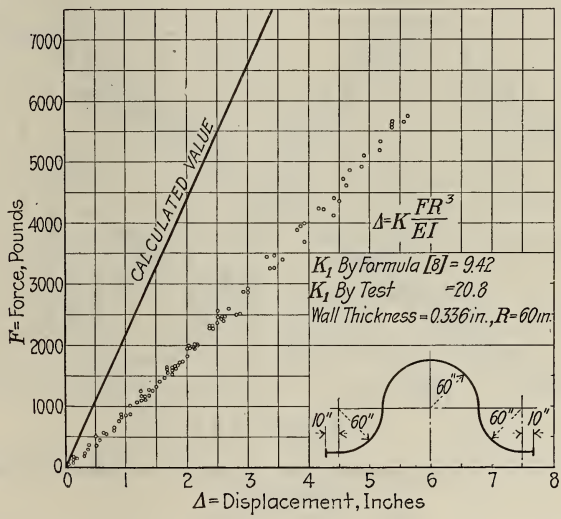


Fig. 17 Force vs. Displacement of a 10-in. Expansion U-Bend — Test Results

port and hold in place, the line of action of the force passing through the pipe to which the bend was connected. All pressure gages were calibrated on a gage tester before being used. Low-reading gages were used for the small forces to insure greater accuracy.

54 Fig. 16 is a photograph showing the jack with the ram extended against the bend. The 6-in. scale shown above the flange was for photographic purposes only and was not used during the tests. Fig. 15 is the same set-up in the released position.

55 In calculating the readings taken, the frictional force was subtracted each time from the total force acting, as given by the corrected gage pressure. This gave the true force causing deflection. The total deflection was obtained by subtracting the corresponding tape reading from the reading when in the released position.

RESULTS OF TESTS

56 In all, eight separate expansion bends were tested. The results are given in detail in Figs. 17 to 24, inclusive, and in condensed form in Table 2.

57 As noted before, the results from bends formed to a radius greater than 5 pipe diameters agreed closely with those obtained

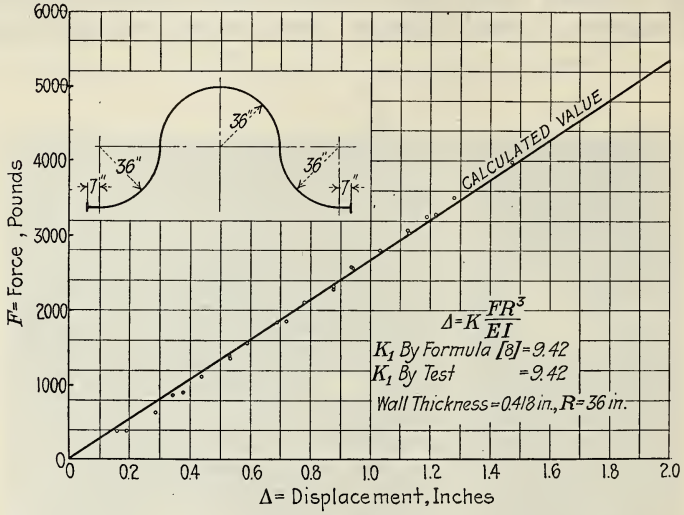


FIG. 18 FORCE VS. DISPLACEMENT OF A 6-IN. EXPANSION U-BEND — TEST RESULTS

TABLE 2 RESULTS OF TESTS BY AUTHORS ON EXPANSION BENDS

Type of U-Bend	Nominal size of pipe, in.	Measured thickness of pipe, in.	Radius of bend, in terms of nominal pipe diameter D	Flexibility index[1] K	
				Calculated	Observed
Expansion U-bend	10	0.336	6D	9.42	20.8
Expansion U-bend	6	0.418	6D	9.42	9.42
Expansion U-bend	6	0.27	5D	9.42	14.08
Expansion U-bend	6	0.274	5D	9.42	16.88
Expansion U-bend	6	0.285	5D	9.42	12.02
Double-offset exp. U-bend	6	0.41	6D	39.88	39.9
Double-offset exp. U-bend	4	0.237	7D	39.88	40.7
Expansion U-bend	3-1/4-in. boiler tube	0.12	8.31D	9.42	9.42

[1] In formulas in Table 1

from the formulas with one exception, this exception being the 10-in. expansion U-bend shown in Fig. 17. The radius in this case was 6 diameters. The pipe in this bend was the largest size tested, all others being 6-in. or smaller. It is relatively more difficult to bend large pipe than small pipe, so as the pipe size increases the minimum allowable radius (expressed in pipe diameters) also has to be increased. One American manufacturer states that the shortest radius to which a 22-in. standard pipe can be bent is 6 diameters,

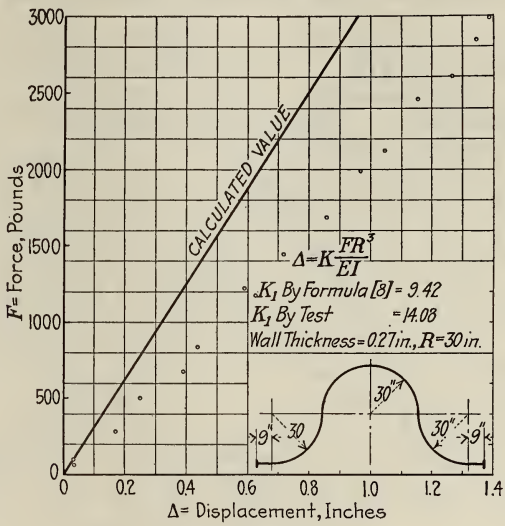


FIG. 19 FORCE VS. DISPLACEMENT OF A 6-IN. EXPANSION U-BEND — TEST RESULTS

while a 5-in. pipe can be bent to 4 diameters. In other words, there is relatively greater danger of the pipe buckling or having waves and folds in it in the larger sizes. It is entirely possible that these are responsible for the unexpected flexibility in this 10-in. bend. The bend was too large to be made from one length of pipe, so it was welded at the top of the loop. The bend was one purchased especially for test and had not seen service.

58 In Figs. 17 to 24 the dots represent actual test points, the force as measured by the gage on the jack being plotted against the displacement of the free end of the bend from its original position. The straight line marked "Calculated Value" is force vs. displacement from a solution of the formula. In solving the formulas the moment of inertia as calculated from the actual measured

thickness of the pipe wall was the one used. The pipe wall was measured with special calipers at several points at each end of the pipe, and the average value used in the calculations.

59 The expansion U-bend shown in Fig. 18 was one purchased especially for test and had not previously seen service. In this case there is a very satisfactory agreement between test and formula.

60 The expansion U-bends shown in Figs. 19, 20 and 21 were bought for the auxiliary superheated-steam line at the Connors Creek power house and had not seen service at the time of test.

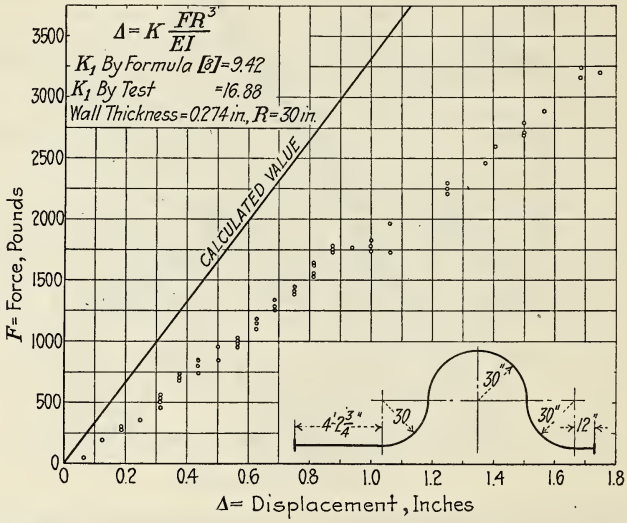


Fig. 20 Force vs. Displacement of a 6-in. Expansion U-Bend — Test Results

All were formed to a radius of 5 diameters and all had a flexibility in excess of the calculated value.

61 The double-offset expansion U-bends shown in Figs. 22 and 23 were purchased especially for test and had a flexibility agreeing closely with the calculated values.

62 In order to eliminate as much as possible the formation of waves or folds in the pipe during the bending operation, the 3¼-in. expansion U-bend shown in Fig. 24 was made up from a straight boiler tube in a Babcock and Wilcox boiler-tube-bending machine. In this machine one end of the tube is clamped to one end of a quadrant having a grooved face and a radius of 27 in. This quadrant turns on its axis and is geared to a motor, the tube sliding against

a stationary grooved wheel on the machine as the quadrant is turned. The length of the tangent from the grooved wheel to the circumference of the quadrant is relatively short. A second grooved wheel on the end of a lever which is pivoted near the axis of the quadrant assists in the bending. In this way compression on the inner side of the bend is minimized, reducing the danger of having waves or folds formed in the tube. The bend was formed cold in this machine, as nearly as possible to shape, and then removed and

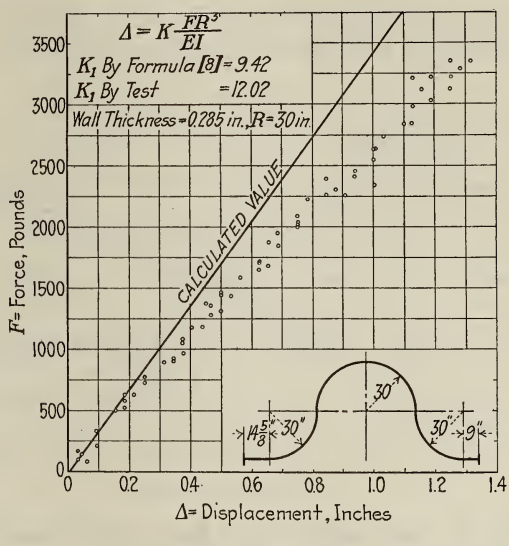


Fig. 21 Force vs. Displacement of a 6-in. Expansion U-Bend — Test Results

heated where necessary, and the bending operation completed. The test results, which are given graphically in Fig. 24, show a very close agreement with calculated values. It should be noted that the radius of the bend with respect to the diameter of the pipe is greater than with any of the other bends tested. This tends to bear out the theory that those bends which are more flexible than the formula would indicate, owe their extra flexibility to waves formed on the compression side of the bend when a pipe is bent to a short radius.

63 Attention is called to the fact that in no case did the tests show a bend to be less flexible than the formulas would indicate, so if the formulas and the design graphs presented in this paper

are used in practice, the designer will always be on the safe side. His bends may or may not be more flexible than his calculations would indicate, depending upon how the bends were made, but he need not fear that forces or stresses in excess of his calculated values will exist.

64 All of the bends were tested while cold. Since the physical properties of mild steel are not altered appreciably up to 700 or 750 deg. fahr., which are about the maximum temperatures to be expected in a steam line with present practice, the relation be-

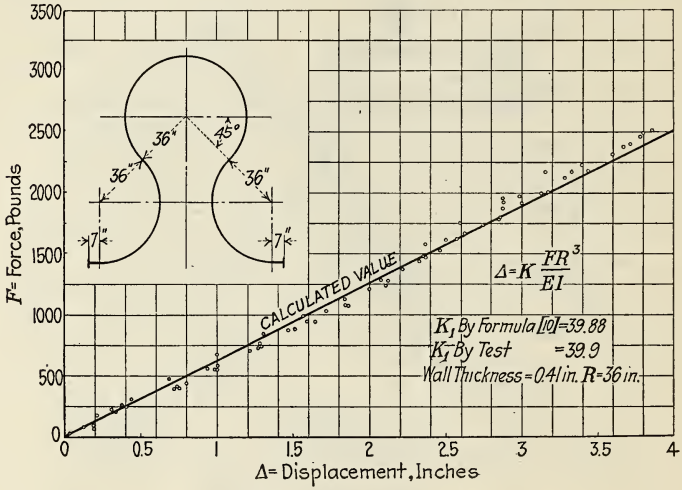


FIG. 22 FORCE VS. DISPLACEMENT OF A 6-IN. DOUBLE-OFFSET EXPANSION U-BEND — TEST RESULTS

tween force and deflection should be the same whether the pipe is hot or cold.

65 The test method used was a rather crude one, yet in several cases the agreement between calculated and test values is remarkably close. It will be noted that in some cases there is a tendency for the force to become disproportionately larger as the displacement increases. This was doubtless due to the binding of the jack as the flange tended to swing to one side, and the flanges to assume positions at an angle to the pipe line. Considering the problem on a purely academic basis, there should be a hinge or knife-edge connection at both flanges in the test set-up, in order to eliminate the errors in testing due to this movement of the flanges; then, so long as the line of action of the force passes through AB,

Figs. 5 and 6, the results should agree with the formulas if the deflection is small and the inner side of the bend is free from waves or folds in the pipe. It should also be noted that in deriving the formulas no allowance was made for a change in shape of the bends as they take up expansion. Actually, the height of the bend, and therefore its flexibility, increases as the bend is deformed, so that for large deflections the actual force will be less than the calculated force. However, as the deflections are relatively small in actual practice, this will not affect the results seriously.

66 In checking a piping layout to determine if excessive fiber stresses exist anywhere in the pipe wall, the longitudinal bursting

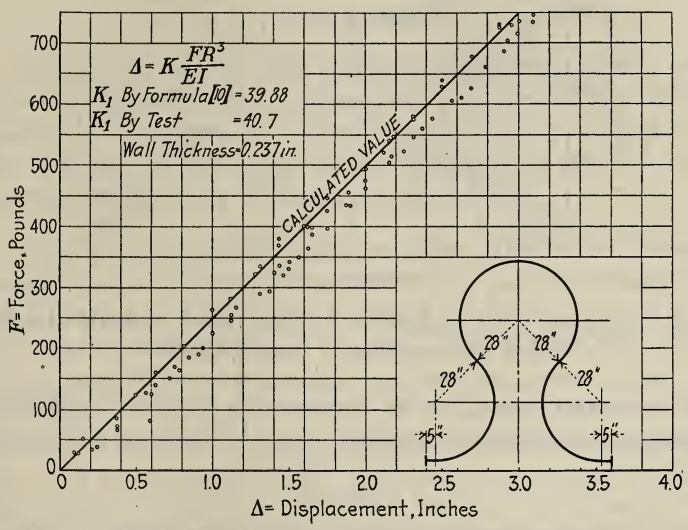


Fig. 23 Force vs. Displacement of a Double-Offset Expansion U-Bend Made from 4-in. Standard Pipe — Test Results

stress due to the pressure of the steam should be added to the bending stress given by the formulas. This is especially important when working with high steam pressures.

67 Nothing has been said about the expansion which can be compressed back into the pipe by the forces set up at the anchorages. In each case it has been assumed that the expansion bend takes care of all of the expansion and that none of it is compressed back into the straight pipe. Actually, of course, a small part of the expansion is taken up by compression, the amount being directly proportional to the length of the pipe and to the force set up, and

inversely proportional to the cross-sectional area of the metal in the pipe and to the modulus of elasticity for that metal. In the case of expansion U-bends and double-offset expansion U-bends, which are very flexible, the amount of expansion compressed back into the pipe is so small compared to the amount taken up by the bend that it can be neglected altogether. In the case of U-bends and quarter-bends, however, the forces are larger and consequently more of the expansion is taken up by compression. In actual prac-

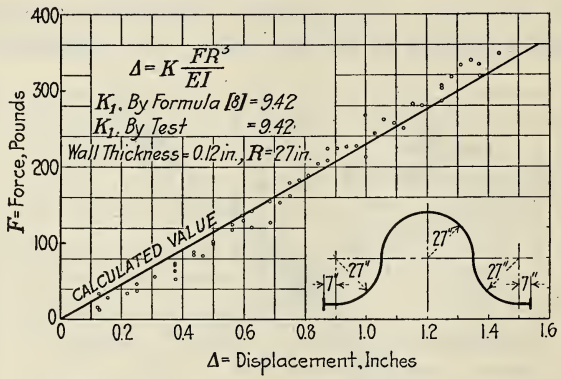


FIG. 24 FORCE VS. DISPLACEMENT OF AN EXPANSION U-BAND MADE FROM $3\frac{1}{4}$-IN. BOILER TUBE — TEST RESULTS

tice the amount taken up by compression may be neglected, for in doing so the designer is making a small allowance on the side of safety.

SUMMARY

68 To summarize in closing: This paper treats of expansion bends in pipe lines as if they were beams acted on by the forces set up by expansion of the piping. Upon this basis formulas have been derived for the forces and stresses in some of the more common types of bends. In order to check the formulas, several bends were tested by applying a force to one end of the bend and then measuring the force and the deflection. In some cases the results substantiated those given by the formulas, and in others the bends were more flexible than the formulas indicated. For an explanation of this, the authors refer to Professor Bantlin's experiments, which tend to show that this extra flexibility present in some cases is due to waves or folds formed in the pipe wall when the bend is made. The formulas are presented graphically in three charts for use in

selecting the proper shape and size of bend to take up a given expansion. In the light of the material presented in this paper the limits of safe expansion values for pipe bends set by Reference (*D*), and commonly quoted in engineering texts and handbooks, are apparently correct only for the double-offset expansion U-bend. For the other shapes the published values are much in excess of those given here. It is the authors' contention that these limits should be revised to agree with the formulas presented in this paper, for unless the published tables are used with large factors of safety, the forces and stresses existing in pipe lines will be excessive and dangerous.

DISCUSSION

George A. Orrok said that he understood all the research had been made with cold pipe without pressure on the inside. He wondered if there was any difference between the action of the pipe bend with two or three hundred pounds pressure on the inside and that of one which was cold and unstressed.

J. A. Freiday, speaking as a representative of Thomas E. Murray, Inc., said that they considered the paper a valuable contribution bearing as it did upon a subject which had not always been given proper consideration in the past. The subject was one which they had carefully considered in the numerous power stations they had designed. Up to two or three years ago they had made use of expansion loops made of pipe bends similar to those described in the paper. In a few cases, he said, they had experienced trouble with bends which evidently had not been properly annealed and which showed a tendency to straighten out and grow after being placed in service. In one case the growth of a 14-in., 90-deg. bend exceeded 2 in. in a few months, the increase being measured after the pipe had been taken out of the line.

In order to eliminate the possibility of such distortion and also to take care of the increasing steam temperatures in larger steam lines required by larger modern units, they had made an investigation some time ago to determine the best method of providing for expansion with the least reaction at the point of anchorage. The result of the investigation, he said, showed that loops corresponding to the expansion U-bends but made up of straight pipe and fittings produced less reaction than an expansion U-bend requiring the same space.

This method of taking care of bends, he said, lent itself particularly to large pipe, for the use of large expansion U-bends was not practical due to fabrication difficulties. With the design of loop they had used in all recent installations no excessive pressure drop had been encountered due to this method of making bends.

He asked the authors if they had investigated any loops designed as he had described.

In making a comparison between the bend made up of straight pipe and fittings and the expansion loop bend, the following formulas were used. For the straight pipe type, the formula for the deflection of bends in Modern Frame Structures, Part 1, by Johnson, Bryan and Turneaure, was used. For the circular bends, the formulas used have been taken from the article by J. G. Stewart published in *Power*, May 10, 1921.

The calculations are based on the assumption that the ends are free to move, which necessitates locating anchors some distance from the bends. In making use of ells for the right angle turns, the deflection will be slightly reduced, but allowing for this it can easily be seen that a bend of this type is more flexible than the standard U-bend. The fiber stress in the ells will tend to increase on account of the short radius but as the body of the fittings is considerably heavier than the body of the pipe it can easily be proven that there is no possibility of rupture at these points.

The comparative formulas[1] follow:

If D = deflection
A = radius of bend in inches
E = modulus of elasticity of material
S = maximum fiber stress, lb. per sq. in.
d = outside diameter of pipe in inches

then for the plain U-bend, Fig. 25,

$$D = \pi \frac{a^2 S}{Ed}$$

But for the straight-line bend using practically the same space

$$D = \frac{16}{3} \frac{a^2 S}{Ed}$$ which is 70 per cent greater.

For the expansion bend, Fig. 26,

$$D = 3\pi \frac{a^2 S}{Ed}$$

[1] See also authors' closure.

and for the straight-line bend

$$D = \frac{40}{3}\,\frac{a^2S}{Ed} \text{ which is 41.5 per cent greater.}$$

For the double expansion bend, Fig. 27,

$$D = 18.48\,\frac{a^2S}{Ed}$$

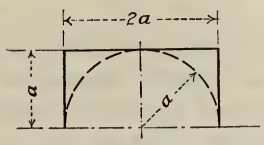


Fig. 25

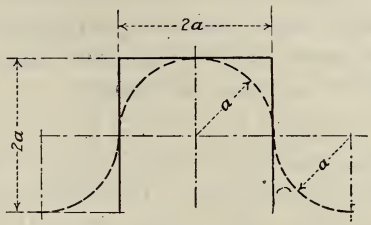


Fig. 26

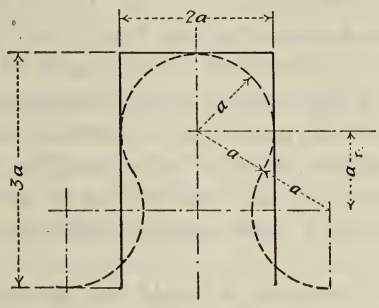


Fig. 27

Figs. 25, 26 and 27 Comparison of Square Loops and Pipe Bends

and for the straight-line bend

$$D = 24\,\frac{a^2S}{Ed} \text{ which is 30 per cent greater.}$$

Francis Hodgkinson. I regard this paper as very valuable and timely. It is, I believe, the first time that concrete information concerning the flexibility of piping has been brought together. The question of providing the requisite degree of flexibility in steam lines is one of prime importance as regards the satisfactory operation of the power units and one which frequently does not receive proper attention. The matter assumes still greater magnitude with modern turbine practice, the capacities of turbines, and hence the physical dimensions of their piping, increasing without corresponding increase in the structure of the turbines themselves. Generally, piping may be planned to have straight runs of pipe at right angles

to each other, and so secure a flexibility without the use of the elaborate bends described in the paper.

A valuable addition to the paper would have been some discussion of resistance of piping to torsional stresses, as it is sometimes difficult to arrange piping to give flexibility in every direction without recourse to its torsional elasticity.

The general arrangement of steam piping to power units demands careful thought, not only to provide against expansion of the piping due to heat, but that small misalignment of flange may be compensated for in every direction.

Robert Cramer said that it was gratifying to learn that the authors had made an attempt to compare directly theoretical and experimental results. He noted in the paper, however, a point of theoretical nature which should be considered when test results were compared with calculated results. In the diagram giving the basis of the theoretical investigation, Figs. 1 and 2, the neutral fiber of the cross-section of the bend was assumed to pass through the center of the circle which was not true in the case of a pipe bend. It was a well-known fact, he said, that the neutral fiber in curved beams did not pass through the center of gravity of the cross-section as it did in straight beams, and the peculiarity was more pronounced the shorter the radius of the beam compared with the magnitude of the cross-section.

For this reason, he said, it would pay the authors to pursue a line of more rigid theoretical investigation of the deviation of the experimental results from the theoretical results in the case of the shorter bends. He said that he noted in making comparison between the experimental and calculated results that in some cases the two checked closely while in others the variation was very marked. It was not the slight variation to be expected from experimental or observational errors but one which would suggest that the authors should look for some error in their calculations.

The authors considered the deflecting force always to be at right angles to the flange on which it was applied with the exception of the case of the quarter bend, in which they had distinguished between two cases, one with the force acting at right angles to the flange and the other with a force acting in the plane of the flange. In the actual case, he said, the problem was really more complicated than would appear from the authors' investigation as it was frequently necessary to twist the flange in order to make a tight joint and this involved another source of strain.

William B. Campbell. The right-hand half of the authors' Table 1 gives data for direct comparison of the calculated relations between force F acting on the bend and the deflection Δ produced with the same relation from actual test. It would appear, however, that the left-hand portion is semi-experimental, in that their calculated values show a relation deduced analytically between the observed deflection Δ and the maximum fiber stress S, while in the line giving "test" results, S is manifestly not observed directly, but is inferred from the experimental value of F which accompanies a given Δ. This method of course reflects the observed relation between deflection and force acting, but it does not eliminate the effect of the method used for calculating S from F, which is based on ideal conditions in the shape and physical dimensions of the pipe. For this reason the "experimental" data in this section are not so conclusive as those in the right-hand section.

H. LeRoy Whitney said he had noticed many examples of square bends and U-bends which did not take up the expansion they had been figured for and also, particularly in long transmission lines, he had noticed the double expansion U-bend had taken over twice the expansion figured for it and that over a period of years no marked deterioration in the pipe was shown. In regard to the buckling of pipe on the inside of the bend, he said that it was perfectly evident that if the pipe did buckle on the inside the fibers on the outside were not stressed nearly as much as if the pipe did not buckle. In large bends of comparatively small thickness it was almost impossible to make them without buckles on the inside. The buckles did no harm, he thought, and they did not retard the flow of steam through the pipe to any appreciable extent and did take up expansion. If the pipe was subject to very high vibrations, he said, it was sometimes customary to pull or push them to the full extent that they would be distorted by expansion so that when they were hot they would be in their normal state relieving the thrust on the steam apparatus.

S. Timoshenko. The authors state that in some of the bends tested, the elasticity found by test exceeded the computed elasticity. Following the explanation given by A. Bantlin,[1] the authors attribute the large deflexions to waves or folds upon the compressed side of the pipe.

[1] *Zeitschrift d. Vereines Deutscher Ingenieure,* 1910, p. 43. See also: *Mitteilungen über Forschungsarbeiten,* Heft 96.

In the following, it is shown that the discrepancy between test results and calculations can be removed by taking account of the distortion of the cross-section of the pipe bends due to bending.[1]

Consider an element of the bend between two neighboring cross-sections ac and bd (Fig. 28). It is easy to see, Fig. 28, that the tensile forces P at the outer side of the bend, and the compressive forces Q at the inner side give resultants T in the direction towards the neutral axis. As a result of this, the previously circular cross-sections become elliptical. Consider now what will be the effect of this distortion on the deformation of the longitudinal fibers of the bend. Let θ denote the initial small angle between the cross-sec-

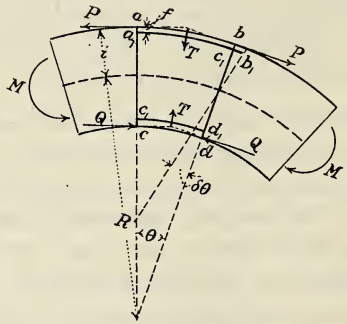


Fig. 28 Element of Bend

tions taken, $\delta\theta$ the small change of this angle at bending. Let the outer fibre ab after bending take the position a_1b_1 and its small displacement towards the neutral axis be denoted by f. The extension of the fiber is

$$l = a_1b_1 - ab = a_1b_1 - a_1c_1 - (ab - a_1c_1) \quad . \quad . \quad . \quad . \quad [\mathbf{a}]$$

Now, assuming that the neutral axis passes through the center of the cross-section, we obtain

$$a_1b_1 - a_1c_1 = c_1b_1 = r\delta\theta$$
$$ab - a_1c_1 = (R + r)\theta - (R + r - f)\theta = f\theta$$

[1] The problem of the distortion of the cross-section of circular pipes as solved by Th. Kármán. See his paper, Uber die Formänderung dünnwandiger Rohre, insbesondere federnder Ausgleichrohre. *Zeitschrift d. Vereines Deutscher-Ingenieure*, Bd. 55 (1911), p. 1889. From this paper are taken the Formulas [**20**] and [**23**] below. The case of a rectangular cross-section of curved pipes is considered in the author's paper to be presented at Spring Meeting of the Society to be held in Montreal, Canada, May 28–31.

Substituting this in [a] and dividing the extension l by the initial length of the fiber ab, equal to $(R + r)\theta$, we obtain the unit elongation of the fiber as follows:

$$e = \frac{l}{(R + r)\theta} = \frac{r}{R + r} \cdot \frac{\delta\theta}{\theta} - \frac{f}{R + r} \quad \ldots\ldots \quad [b]$$

The first member on the right represents the unit elongation due to the rotation of the cross-section bd in respect to the cross-section ac. The second member on the right side gives us the effect of the distortion of the cross-section. It is easy to see that this effect is very substantial. If we take, for instance, $R + r = 60$ in., and put $f = 0.02$ in. only, we obtain

$$\frac{f}{R + r} = \frac{1}{3000}$$

The corresponding stress is

$$E \cdot \frac{f}{R + r} = \frac{3.10^7}{3.10^3} = 10000 \text{ lb. per sq. in.}$$

We see that a very small displacement f produces substantial diminishing of the stress in the fiber ab.

Similar conditions exist also on the opposite side. The compression of the fiber cd will be diminished by the displacement of the fiber toward neutral axis.

As a result of the distortion or rather the flattening of the cross-section the outer fibers do not share in the stress distribution as required by the ordinary theory of flexure. In respect to the deformation, the flattening of the cross-section has an effect as if the moment of inertia had been diminished. That is, if I is the moment of inertia we have to use in its place the value κI in the equation of flexure of a curved bar; i.e. instead of the equation

$$M = EI \frac{\delta\theta}{R\theta} \quad \ldots\ldots\ldots\ldots \quad [18]$$

we will use the equation

$$M = \kappa EI \frac{\delta\theta}{R\theta} \quad \ldots\ldots\ldots\ldots \quad [19]$$

Where κ is a coefficient, smaller than unity, which is to be calculated from the formula.[1]

$$\kappa = 1 - \frac{9}{10 + 12\left(\frac{tR}{r^2}\right)^2} \quad \ldots\ldots\ldots \quad [20]$$

[1] See the paper of Th. Kármán, cited above.

Here t is the thickness of the tube, and $r = \frac{D - t}{2}$, where D is the outer diameter of the tube.

We see that the effect of the flattening of the cross-section on the flexibility of the bend depends on the ratio $\frac{tR}{r^2}$ only.

With decreasing of t and R and with increasing of the radius r of the tube, the coefficient κ decreases, i.e., the flexibility of the bend and its deflexion increase.

From [**18**] and [**19**] it follows that the true deflexion can be obtained by multiplying the deflection, obtained from [**18**], by κ.

Now we shall apply the above to those cases where authors found the greatest deviation of the experimental results from the theoretical results.

For the tube of 10 in. with $R = 60$ in., Equation [**18**] gives for the deflexion $\Delta = 9.42$ in. The experiment gave for the same case the deflexion 20.8 in. i.e. 121 per cent greater.

Now, from [**20**] we obtain.

$$\kappa = 0.525$$

and the corresponding deflexion is

$$\Delta_1 = \frac{\Delta}{\kappa} = \frac{9.42}{0.525} = 18.0 \text{ in.}$$

which differs about 16 per cent from the experimental datum. This result may be considered as satisfactory, especially if we take into account the probable error in the measurement of the thickness t and the fact that the modulus E was taken equal to 3.10^7 lb. per sq. in. instead of being determined by experiment.

For the tube of 6 in. with $t = 0.274$ in. and $R = 30$ in., the authors obtained from calculations $\Delta = 9.42$ in. and from experiment $\Delta = 16.88$ in. For this case $\kappa = 0.585$ and the deflexion as given by Equation [**19**] is

$$\Delta_1 = \frac{9.42}{0.585} = 16.1 \text{ in.}$$

which is in very good coincidence with the experiment.

In order to obtain from the design graph, given by the authors, the true values of the deflexion Δ_1, and of the forces F_1, acting against anchorage, it is necessary only to calculate from [**20**] the corresponding κ. Then

$$\Delta_1 = \frac{\Delta}{\kappa}, \qquad F_1 = \kappa F \qquad \ldots\ldots\ldots \quad [\mathbf{21}]$$

where Δ and F are the quantities given by the graphs.

Consider now the stresses. Taking into account the distortion of cross-sections, it is necessary to replace the usual formula for stresses.

$$S = \frac{My}{I} \quad \ldots\ldots\ldots\ldots \quad [22]$$

by the more complicated formula [1]

$$S = \frac{My}{\kappa I}\left(1 - \beta\frac{y^2}{r^2}\right) \quad \ldots\ldots\ldots \quad [23]$$

where in

$$\beta = \frac{6}{5 + 6\left(\frac{tR}{r^2}\right)^2} \quad \ldots\ldots\ldots \quad [24]$$

For the maximum stress, we obtain now, instead of

$$S_{max} = \frac{MD}{2I} \quad \ldots\ldots\ldots\ldots \quad [25]$$

the value

$$S_{max} = \gamma \cdot \frac{MD}{2I} \quad \ldots\ldots\ldots\ldots \quad [26]$$

where

$$\gamma = \frac{2}{3\kappa\sqrt{3\beta}} \quad \ldots\ldots\ldots\ldots \quad [27]$$

Some values of this coefficient are given below:

$\frac{tR}{r^2} =$	0.3	0.5	1.0
$\gamma =$	1.98	1.30	0.88

We see that in cases where $\frac{tR}{r^2}$ is small, the real maximum stress is materially greater than follows from the formula [8]. In calculating the maximum stresses from the design graphs given, it is necessary to take account of the fact that, as result of greater flexibility the forces and the bending moments are diminished in the ratio $\kappa : 1$. Therefore, the true values of maximum stress are obtained by multiplying those given by the graphs, by $\kappa\gamma$. For the cases of the tube of 10 in. and 6 in., considered above, we obtain $\kappa\gamma = 0.607$ and $\kappa\gamma = 0.658$ respectively, i.e., the true value of the maximum stress is about 40 and 35 per cent less than it is given by the graphs.

The Authors are indebted to those who have contributed several valuable suggestions in the discussion of this paper, and they are glad of this opportunity to amplify their explanation of certain points which evidently were not sufficiently covered in their original work.

[1] See the paper of Th. Kármán, cited above.

The point raised by Robert Cramer in regard to the displacement of the neutral axis in a radial cross-section of a bent beam fails to explain the observed variation between calculated and test results. This displacement of the neutral axis tends to increase the moment of inertia of the section, thus giving the bend a slightly greater calculated resistance to deflection. The variation then is somewhat worse than when the neutral axis is taken through the center of the circle. However, this displacement does have a bearing on the calculated results as explained in the following paragraphs. Since the authors do not know of any published calculations in convenient form for use, they have included in detail their derivation for this displacement, and have shown wherein these calculations slightly modify the formulas and curves of the paper.

LOCATION OF NEUTRAL AXIS

PIPE BEND CONSIDERED AS INITIALLY CURVED BEAM

In a pipe bend of radius R (see Fig. 29) the neutral axis $N'-N'$ of any radial cross-section is situated at a distance h measured toward

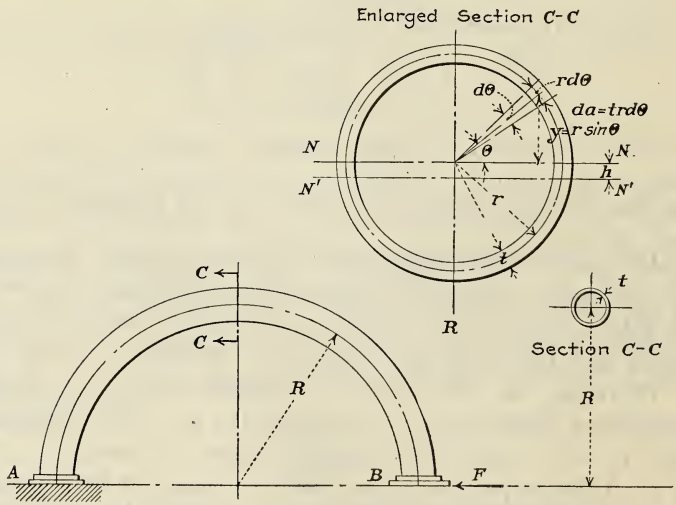


FIG. 29 LOCATION OF NEUTRAL AXIS

the center of curvature from the central axis $N-N$. Strength of Materials by Morley, Chapter 12, Articles 129 and 130, gives the following general formula for the distance h in curved beams:

$$h = -R\frac{A' - A}{A'} \quad \text{[28]}$$

where $A' = R\Sigma\left(\frac{da}{y + R}\right)$

da = element of area
y = vertical distance from central axis to element of area
$A = \Sigma da$ = area of section.

Applying the formula for A' to a pipe bend (see Fig. 29) then:

$$da = trd\theta \qquad y = r \sin \theta$$

$$\tfrac{1}{2}A' = R \int_{\theta = -\frac{\pi}{2}}^{\theta = \frac{\pi}{2}} \frac{trd\theta}{r \sin \theta + R}$$

$$A' = 2Rtr \int_{\theta = -\frac{\pi}{2}}^{\theta = \frac{\pi}{2}} \frac{d\theta}{r \sin \theta + R}$$

From Peirce's Table of Integrals, placing $Z = \tan \frac{1}{2}\theta$ and changing the limits to agree with the new variable:

$$A' = 2Rtr \int_{Z=-1}^{Z=1} \frac{2dZ}{R + 2rZ + RZ^2}$$

$$= 4Rtr \left[\frac{2}{\sqrt{4R^2 - 4r^2}} \tan^{-1} \frac{2RZ + 2r}{\sqrt{4R^2 - 4r^2}} \right]_{Z=-1}^{Z=1}$$

$$= \frac{4Rtr}{\sqrt{R^2 - r^2}} \left[\left(\tan^{-1} \frac{R + r}{R^2 - r^2} \right) - \left(\tan^{-1} \frac{-R + r}{\sqrt{R^2 - r^2}} \right) \right]$$

$$= \frac{4Rtr}{\sqrt{R^2 - r^2}} \left[\left(\tan^{-1} \frac{R + r}{\sqrt{R^2 - r^2}} \right) + \left(\tan^{-1} \frac{R - r}{\sqrt{R^2 - r^2}} \right) \right]$$

$$= \frac{4Rtr}{\sqrt{R^2 - r^2}} \left[\tan^{-1} \sqrt{\frac{R + r}{R - r}} + \tan^{-1} \sqrt{\frac{R - r}{R + r}} \right]$$

$$= \frac{4Rtr}{\sqrt{R^2 - r^2}} \left[\tan^{-1} \sqrt{\frac{R + r}{R - r}} + \left(\frac{\pi}{2} - \tan^{-1} \sqrt{\frac{R + r}{R - r}} \right) \right]$$

$$= \frac{4Rtr}{\sqrt{R^2 - r^2}} \times \frac{\pi}{2} = \frac{2\pi rtR}{\sqrt{R^2 - r^2}} \quad \ldots\ldots\ldots\ldots \quad [29]$$

Substituting in Formula [28] the value of A' from Formula [29] and the value of $A = \Sigma da = \Sigma trd\theta = 2\pi rt$

$$h = -R \frac{\dfrac{2\pi rtR}{\sqrt{R^2 - r^2}} - 2\pi rt}{\dfrac{2\pi rtR}{\sqrt{R^2 - r^2}}} = -(R - \sqrt{R^2 - r^2}) = \sqrt{R^2 - r^2} - R. \quad [30]$$

See Fig. 30 for numerical values.

If the moment of inertia of section $C-C$ about $N-N$ is designated as I_N and about $N'-N'$ as $I_{N'}$ then:

$$I_{N'} = I_N + Ah^2$$

The sharpest curvature ordinarily encountered in American commercial practice occurs in pipes bent to a radius of five pipe diameters. A solution for $I_{N'}$ in general terms covering all pipes bent to a radius of five diameters gives $I_N = 1.005I_N$, or in other words the moment of inertia is increased by $\frac{1}{2}$ of 1 per cent in bent

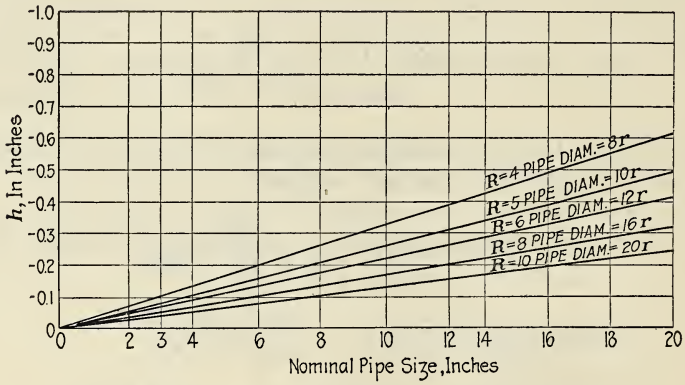


FIG. 30 RELATION OF h AND NOMINAL PIPE SIZES

$h = \sqrt{R^2 - r^2} - R$
R = radius of pipe bend
r = mean radius of pipe
Sizes 2 in. — 12 in. are standard weight pipe
Sizes 14 in. — 20 in. are O.D. pipe — $\frac{3}{8}$in. thick

Maximum fiber stresses of Fig. 12 are to be increased by percentage $\frac{h}{r}$ 100 per cent.

r must be taken as mean radius of pipe and not as nominal radius.

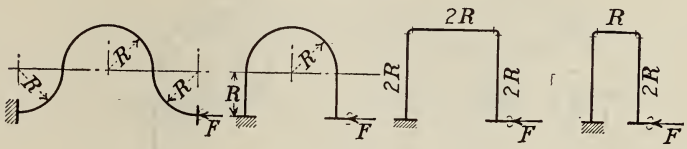
pipes having the sharpest curvature to be expected. With a larger ratio of R to r the value of $I_{N'}$ will more closely approach I_N. Hence in figuring anchor thrusts, which are directly proportional to the moment of inertia, it would appear sufficiently accurate for practical purposes to neglect the effect of curvature on the location of the neutral axis, and assume that it passes through the center of the section.

However, with a radius of five pipe diameters, which also is the worst condition in regard to stress, the maximum fiber stress is increased nearly 5 per cent. Since it is obviously impracticable to complicate the stress formulas [5], [7], etc., by introducing an expres-

sion involving the value of h, this increased stress may be taken into account, at the discretion of the designer, by adding a percentage to the stress values worked out from the stress formula, or taken from Fig. 12. Fig. 30 furnishes a convenient means of determining the percentage to add, which is approximately $\frac{h}{r} \times 100$ per cent. It should be noted that r must be taken as the mean radius of the pipe and not the nominal radius.

COMPARISON OF SQUARE LOOPS WITH PIPE BENDS

J. A. Freiday asks for information in regard to the flexibility of square loops made up of straight pipe and fittings instead of bent



$\Delta = 9.42 \frac{FR^3}{EI}$	$\Delta = 9.38 \frac{FR^3}{EI}$	$\Delta = 13.33 \frac{FR^3}{EI}$	$\Delta = 9.33 \frac{FR^3}{EI}$
$S = \frac{D\Delta E}{9.42R^2}$	$S = \frac{D\Delta E}{9.38R^2}$	$S = \frac{D\Delta E}{13.33R^2}$	$S = \frac{D\Delta E}{9.33R^2}$
$l = 6.2832R$	$l = 5.1416R$	$l = 6R$	$l = 5R$

l = Developed length of pipe

FIG. 31 A COMPARISON OF SQUARE LOOPS AND PIPE BENDS

pipe. The authors have not tested any loops of this kind, but the formulas worked out in their paper and given in connection with Figs. 7, 8 and 9 apply to such loops. Fig. 31 has been added to furnish a comparison between square loops and pipe bends occupying about the same space. An obvious disadvantage of the square loops is that they have four more joints per loop than a pipe bend (provided the bend can be made from one piece of pipe) and that they introduce two sharp 90-deg. elbows in place of one easy 180-deg. bend.

EFFECT OF INTERNAL PRESSURE AND TEMPERATURE

George A. Orrok raises a question regarding the effect of internal pressure on the pipe walls. In figuring maximum stresses in actual piping designs the authors have made it a practice to combine the maximum fiber stress due to bending with any other stresses which are of sufficient magnitude to be considered. These stresses might include, depending on the location of the section of maximum

stress, bursting stress and longitudinal stress due to internal pressure, shearing stress, and direct compressive stress due to anchor thrust.

The authors believe that the temperature at which the pipe is operated would not immediately affect the stress and thrust conditions, provided the maximum combined stress does not exceed the elastic limit of the material at the temperature in question. Gradual distortion or growth of the metal, while subject to high temperatures over a long period of time, may eventually produce a change from the original conditions of stress and thrust. Instances of this nature have been observed in the superheated steam lines of The Detroit Edison Company, when pipes which were originally

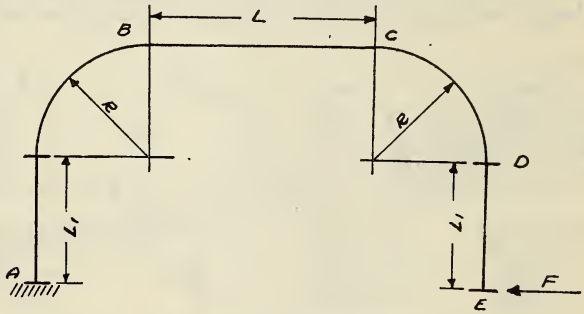


Fig. 32 Special U-Bend Made from Quarter Bends and Tangents

cold-sprung did not return to their original position when unbolted after a period of service. For this reason the authors doubt whether it is safe practice to rely too much on cold springing for relieving expansion forces in high temperature lines.

U-BEND MADE FROM QUARTER BENDS AND TANGENTS

Consider Fig. 32 with a force F acting along the line AE. Flange A is fixed. Δ is the total deflection of flange E measured in the direction of F. Let Δ_1 be the deflection of flange E due to bending in DE, Δ_2 the deflection due to bending in CD and Δ_3 the deflection due to bending in BC. Then

$$\Delta = 2(\Delta_1 + \Delta_2) + \Delta_3$$

If we consider DE to be of length L_1 and BC of length L, then from Pars. 38, 39 and 40 we obtain the following:

$$\Delta_1 = \frac{FL_1^3}{3EI}$$

$$\Delta_2 = \frac{FR}{EI}\left(\frac{\pi L_1^2}{2} + \frac{\pi R^2}{4} + 2L_1R\right)$$

$$\Delta_3 = \frac{FL(L_1 + R)^2}{EI}$$

$$\Delta = \frac{F}{EI}\left[\frac{2L_1^3}{3} + 2R\left(\frac{\pi L_1^2}{2} + \frac{\pi R^2}{4} + 2L_1R\right) + L(L_1 + R)^2\right]. \quad [31]$$

The maximum fiber stress occurs at the top of the bend

$$S = \frac{F(R + L_1)D}{2I} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; [32]$$

EXPANSION U-BEND MADE FROM QUARTER BENDS AND TANGENTS

Consider Fig. 33 with a force F acting along the line AG. Flange A is fixed. Δ is the total deflection of flange G measured in the

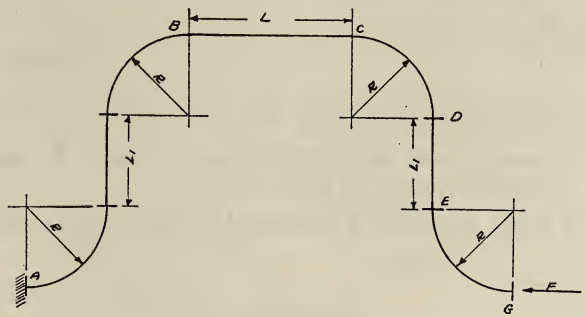


FIG. 33 SPECIAL EXPANSION U-BEND MADE FROM QUARTER BENDS AND TANGENTS

direction of F. Let Δ_1 be the deflection of flange G due to bending in EG, Δ_2 the deflection due to bending in DE, Δ_3 the deflection due to bending in CD and Δ_4 the deflection due to bending in BC.

Then $$\Delta = 2(\Delta_1 + \Delta_2 + \Delta_3) + \Delta_4$$

$$\Delta_1 + \Delta_2 = \frac{F}{EI}\,\frac{(R + L_1)^3 + 0.069R^3}{3} \text{ from formula [12]}$$

and from Pars. 39 and 40:

$$\Delta_3 = \frac{FR}{EI}\left[\frac{\pi(L_1 + R)^2}{2} + \frac{\pi R^2}{4} + 2R(L_1 + R)\right]$$

$$\Delta_4 = \frac{FL(L_1 + 2R)^2}{EI}$$

$$\Delta = \frac{F}{EI}\left\{\frac{2[(R + L_1)^3 + 0.069R^3]}{3} + R\left[\pi(L_1 + R)^2 + \frac{\pi R^2}{2} + 4R(L_1 + R)\right] + L(L_1 + 2R)^2\right\} \quad . \; . \; . \; . \; . \; . \; . \; [33]$$

The maximum bending stress occurs at the top of the bend.

$$S = \frac{FD(2R + L_1)}{2I} \quad \text{[34]}$$

Tangents at A and G are not shown inasmuch as they lie in the line of action of the force and therefore have no effect on the flexibility of the bend.

ELASTICITY OF A STRAIGHT PIPE IN TORSION

Consider the pipe shown in Fig. 35. End A is anchored in such a way that it is not free to turn, and end B is subjected to a torque T.

The fundamental torsion formula is

$$T = \frac{qJ}{r}$$

where

T = torque, inch pounds
q = maximum shearing stress due to torsion, lb. per sq. in.
r = outside radius of pipe, inches
J = polar moment of inertia of circular cross section about axis of pipe.

$$T = FL_1$$

where

F = force, pounds

and

L_1 = distance in inches of line of action of force F from center line of pipe under torsion.

$$G = \frac{\text{shearing stress}}{\text{shearing strain}}$$

G = modulus of shearing elasticity (11,500,000 lb. per sq. in. for steel)

$$\text{Shearing strain} = \frac{r\,\theta}{L}$$

θ = angle of twist
L = length of pipe under torsion, inches.

$$G = \frac{q}{\frac{r\theta}{L}} \qquad q = \frac{r\,\theta\,G}{L}$$

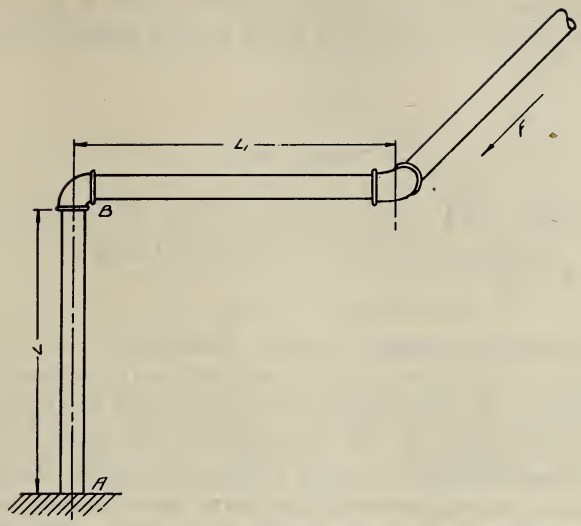


Fig. 34 Arrangement of Pipe Line to Take up Expansion by Torsion

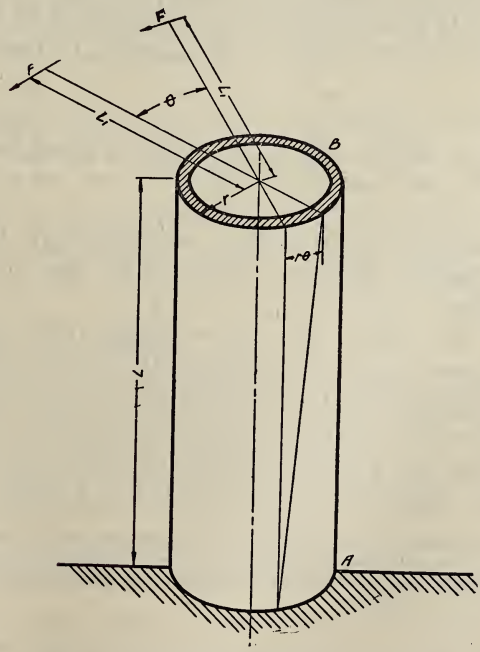


Fig. 35 Straight Pipe in Torsion

Let Δ be the distance through which the force F acts, in other words the expansion of a pipe line which is to be taken up in torsion

$$\Delta = \theta L_1$$

$$q = \frac{r\Delta G}{LL_1} \quad \text{[35]}$$

$$FL_1 = \frac{qJ}{r} = \frac{JG\Delta}{LL_1}$$

$$\Delta = \frac{FLL_1^2}{JG} \quad \text{[36]}$$

The formulas suggested by Dr. Timoshenko [1] do not satisfactorily clear up the discrepancies observed by the authors between their calculated and test results. It is true that in the two cases selected for illustration by Dr. Timoshenko, the use of his formula gives a calculated result very close to the test figures. In the other six cases, however, it gives results differing from the test results by a larger margin than do those calculated from the authors' formula. The authors have solved Dr. Timoshenko's formula to determine his flexibility index K_2 for each of the eight bends tested. The values of his index are tabulated below for comparison with the authors' calculated and observed values of K as given in Figs. 17 to 24.

Figure	17	18	19	20	21	22	23	24
Authors' index K:								
Calculated	9.42	9.42	9.42	9.42	9.42	39.88	39.88	9.42
Observed	20.8	9.42	14.08	16.88	12.02	39.9	40.7	9.42
Timoshenko index, K_2	20.7	12.2	19.2	18.9	18.2	52.3	53.5	13.3

It will be noted that the authors' solution of Dr. Timoshenko's formula gives slightly different values for K_2 than those given in his two examples.

In view of the above, the authors believe that their formula gives results more closely approaching the actual conditions for most pipe bends than does Dr. Timoshenko's. At least the authors' formula is on the side of safety in design, as it shows a larger thrust and higher stress for a given deflection than his does, and hence represents more nearly the worst conditions to design against.

[1] On page 588, lines 14 to 30, where the word "deflexion" occurs, substitute "index"; and where Δ and Δ_1 occur, substitute K and K_2, respectively. After making these substitutions, strike out the abbreviation "in." after all numerical values of K and K_2. Also, in Figs. 17 to 24 inclusive, for K_1 read K. — Editor.

No. 1861

SAFETY ENGINEERING IN THE COMPRESSION OF GASES

By A. D. Risteen, Hartford, Conn.

Member of the Society.

The purpose of this paper is to outline a few of the chief hazards that are associated with the compression of some of the gases in common use in industry. Among the gases considered are air, oxygen, nitrogen, argon, carbon dioxide, hydrogen, acetylene, ammonia, and chlorine. The preparation, utilization, storage, or transportation of the gases, except when some of these items may happen to have an important bearing upon the actual operation of compression, are not discussed. As to the mechanical strength of the apparatus that is used to effect the compression, the paper deals only with the things that are likely to happen even when the apparatus itself is strong enough to withstand the stresses that are thrown upon it in the course of its normal operation.

COMPRESSED air, the first of the gases to be considered, is used for many different purposes and at many different pressures, but so far as safety is concerned it is sufficient to distinguish two main problems. First, we have to deal with cases in which the ultimate pressure desired does not exceed, say, 200 lb. per sq. in.; and second, the special case in which the compression must be pushed to perhaps 3000 lb. per sq. in., for the production of liquid air.

2 When the ultimate pressure is not greater than 75 lb., the compression is usually effected in a single operation; but it would seem better to adopt two-stage compressions for pressures approaching or exceeding this limit. Three stages, at least, should be used for pressures in the neighborhood of 1000 lb., and to push the pressure from this point up to 3000 lb. a fourth stage or operation should be employed. It is advisable to use long-stroke com-

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pressors, with low piston speed, not only to avoid unnecessarily high temperatures, but also because less lubrication is needed.

3 Every now and then there is an explosion in connection with an air compressor, and the results are sometimes quite serious. Most of these explosions are probably associated more or less directly with the lubrication of the compressor cylinders, and with inefficient cooling of the air that is undergoing compression. In an ordinary compressor the temperature of the air rises considerably during the compression stroke, and it is of the utmost importance to keep this rise of temperature within reasonable bounds. A good deal can be accomplished in this direction by surrounding the cylinders with effective water jackets. The cooling water in these jackets should circulate actively and copiously, and thermometers should be provided to show the temperature at which the water is entering and leaving. To guard against stoppage of the flow of cooling water from any cause, it is also important to provide some positive means for showing that the circulation is free and plentiful at all times. It is advisable, for example, to have the discharge from each jacket located where it is plainly visible to the men working about the room.

4 It is not sufficient, however, to provide the compressor cylinders with water jackets. The air should be passed through a special cooler immediately after leaving the compressor and its temperature brought down near that of the surrounding atmosphere as quickly as practicable. Moreover, if the compression is effected in two or more stages, an intercooler between each stage and the next one should be provided, so that every cylinder will be supplied with air at a moderate and reasonable initial temperature. The use of efficient cooling devices tends not only to insure safety but also to lessen the cost of compression, because cooling reduces the pressure as well as the temperature.

5 In compressing air to moderate pressures the cylinders are lubricated with oil. Oil may also be used in the first stages of high-compression apparatus, though water is preferable for the final stages. The composition and physical characteristics of the oil should be carefully considered.

6 As a general rule, too much oil is used in lubricating the cylinders of air compressors. It takes considerable experience to determine just the right amount, and what is right with one machine or one grade of oil may not be right with another. In a

general way, however, it is safe to say that the lubrication is ample if the cylinder walls are always coated with a slight film of oil. It is often assumed that the explosions that sometimes occur in connection with air compressors are due to the ignition of oil vapor or of oily mist. It would be hard to justify this theory. Whenever a compressor cylinder discharges the air that it contains, the oily vapor or spray that is present is discharged at the same time, and it is hard to see how a quantity sufficient to produce a serious explosion could accumulate. Moreover there are comparatively few explosions in which the initial break is in the compressor cylinder. More commonly the first rupture occurs in the discharge pipe or the receiver, and trouble at these points can be minimized, and perhaps entirely avoided, by (1) quick and efficient cooling of the air, (2) thorough drainage of the oil that tends to collect in the piping and receiver, and (3) careful attention to the discharge valves on the compression apparatus. Considerable quantities of oil are likely to accumulate in the discharge pipe and the receiver, and if either of these bursts, a large amount of hot oily spray is likely to be discharged into the air, and this may take fire either spontaneously or from some external source, with a bad oil-vapor explosion as the result. Suitable traps and drains must be provided to prevent the accumulation of the oil, and in this way the oil hazard can be largely or wholly removed.

7 If the exhaust valves of the compression apparatus are leaky, then upon the return stroke more or less of the compressed and heated air will find its way back into the cylinder, so that the temperature of the air in the cylinder upon the next compression stroke will reach a much higher point than the designer of the apparatus intended. Carbonaceous deposits, produced by the carbonization of the lubricating oil, often collect in considerable quantity in and around the exhaust valves, and these may sometimes take fire, or they may prevent the valves from closing tightly. Exhaust valves should be carefully watched for such deposits, and should be kept as clean and tight as possible.

OXYGEN

8 Oxygen is compressed and shipped in vast quantities and is used in the greatest imaginable variety of ways. Before subjecting it to compression it is essential to know that the oxygen

is pure, and especially to know that it is not contaminated with any other substance with which it could combine, either during compression or in the course of subsequent handling. For example, if it is prepared by the fractional distillation of liquid air, it is important to know that the air that was subjected to liquefaction was free from smoke particles, from organic dust of any kind, and from coal gas, acetylene, and every other substance of an oxidizable nature. If the oxygen was produced by the electrolysis of water, it is equally essential to know that it is not contaminated by hydrogen. It is easy enough to insure purity in all these respects, but the importance of not neglecting the necessary testing, and the purification operations when these prove to be needed, cannot be over emphasized.

9 In handling compressed oxygen it is supremely important to prevent the gas from coming in contact with oil or grease either during the compression or while the oxygen is being stored, transported, or utilized; and in view of the fact that many exceedingly serious accidents have occurred in consequence of neglecting this principle, the author wishes to emphasize the hazard just as strongly as he can. When a man talks with appropriate earnestness about the danger of allowing compressed oxygen to come in contact with oil, grease, or other combustible organic materials, it is easy for the uninitiated to believe that he is inspired by unreasonable timidity, but this is not the case. Those who have followed the history of oxygen compression are well aware that many accidents have occurred under conditions that thoughtful and experienced men would not have considered to be at all likely to produce trouble. The use of even a slight trace of oil in the compressor is exceedingly hazardous, not only on account of the danger of immediate combustion, but also because some part of the oil is bound to go over with the compressed gas; this will collect in the tanks or cylinders used for storage purposes, and sooner or later serious results will surely follow. It is probably safe to say that most of the accidents that have occurred in connection with compressed oxygen have been due to lack of appreciation of the importance of this thing. The trouble is not confined, of course, to the mere combustion of the oil itself. The metal parts of the apparatus will burn freely as soon as they become sufficiently heated by the ignition of the oil or grease, and the consequences may transcend imagination.

10 The only absolutely safe thing to use for the lubrication of an oxygen-compressor cylinder is *pure water,* which should be introduced into the suction pipe in the form of a visible spray. Under special conditions it may happen that pure water is not sufficient, and in a case of this kind it may be permissible to dissolve a little soap in it; but soap should not be used when water will suffice, and it should *never* be used without first proving, by actual analysis, that the particular supply to be dissolved contains no uncombined (or unsaponified) oil or grease. The packings used on the compressor must also be entirely free from oil, grease, or graphite. Special packings, made for this express purpose, should be invariably employed.

11 The compression of oxygen is often, and perhaps usually, carried up to about 2000 lb. per sq. in., and it should be effected in not less than three stages. In speaking of the compression of air, the importance of prompt and effective *cooling* has been emphasized, and much greater attention should be paid to this point when compressing oxygen. The pump cylinders should be water-jacketed, the circulation of water should be positive and abundant, and the apparatus should be so designed that cessation or diminution of the flow of jacket water cannot escape immediate notice. The gas should also be passed through effective intercoolers between each compression stage and the one next following; and *immediately* after leaving the last cylinder of the compressor it must be passed through a final cooler, and brought down to the temperature of the surrounding atmosphere.

12 It is exceedingly unwise to introduce oxygen into receivers of any kind, unless these receivers are known to be perfectly clean internally and wholly free from oil, grease, and other combustible substances. The author would like to see some kind of a regulation established forbidding every oxygen manufacturer to fill any tanks other than his own. This principle is already followed, it is believed, by some and perhaps all of the larger manufacturers, but unfortunately it is not followed by everybody. Moreover, oxygen should never be compressed into containers that have been used for other gases. When a cylinder or tank has been previously used for holding compressed gases other than oxygen, there is likely to be an accumulation of oil upon its inner surface, because oil may have been used for cylinder lubrication in connection with the compression of these other gases, and it is hard to remove such oil thoroughly enough to make the tank safe for oxygen.

NITROGEN

13 Nitrogen gas has a far narrower range of application in the arts than oxygen, yet it is compressed and shipped to a certain extent. This gas was used during the war in the recoil cylinders of certain types of guns, and it has also been used to a limited extent for blanketing combustible liquids and for supplying a neutral atmosphere for other purposes, where the presence of a gas with active chemical properties would be objectionable. At the present time compressed nitrogen is supplied mainly, it is believed, for filling incandescent electric lamps, and in connection with the manufacture of automobile tires. The gas itself is coming into use for the direct manufacture of ammonia, nitric acid, fertilizers, and other nitrogen compounds from the atmosphere; but chemical plants in which this kind of work is done usually prepare their own nitrogen and do not subject it to compression except as the process may require it to be under pressure.

14 Nitrogen forms four-fifths of the bulk of the atmosphere, and as it is nearly inert in its elemental form the precautions that are recommended in connection with the compression of air are also adequate when compressing nitrogen. It should be noted, however, that the nitrogen that is used for filling electric-lamp bulbs must be free from hydrocarbons, and this means that when the gas is to be employed for this purpose it is not permissible to use oil for lubricating the compressor cylinders. Water should be used instead. For certain reasons, in fact, it is best to avoid oil altogether and to lubricate with water in all cases. This does not mean, however, that there is any danger of direct chemical action between the oil and the nitrogen.

15 Nitrogen is usually compressed to about 2000 lb. per sq. in., and the compression should be performed in three stages.

ARGON

16 Argon, like helium and neon, has no chemical properties whatsoever. It forms no compounds and is absolutely inert under all circumstances. Hence it cannot produce fires or explosions. Moreover, it is not poisonous. It is obtained from liquid air, however, and when first separated from the air it is mingled with a large proportion of oxygen — the mixture usually containing

about 65 per cent of oxygen and 35 of argon. When it is handled in this state it must therefore be treated like oxygen; but after it has been freed from the admixed oxygen it may be compressed under the same conditions as nitrogen. For shipment it is usually compressed to a pressure of 1800 or 2000 lb. per sq. in., in three stages.

CARBON DIOXIDE

17 This gas is compressed in large quantities and is used for the most varied purposes. As is well known, it will neither burn nor support combustion, and the author does not recall any chemical dangers which are likely to be encountered in its compression. It is not especially poisonous, but when a considerable quantity of it is mixed with the air that the workmen have to breathe, it replaces the life-giving oxygen to a corresponding extent, and this may mean that the air is not fit for respiration. The effects of carbon dioxide in this way are sometimes very insidious, especially as the pure gas is without odor or color; and it is therefore evident that special care should be taken to provide free and copious ventilation at all times in buildings in which carbon dioxide may escape, by leakage or otherwise.

HYDROGEN

18 In connection with hydrogen and most of the other combustible gases (such as propane), the chief danger associated with compression probably consists in the likelihood of fire or explosion in case of leakage or of contamination with oxygen. If the hydrogen is prepared by the electrolysis of water, there is always a possibility that oxygen may be present in it in small amounts, and the gas should be carefully tested, either at short intervals or preferably by some continuous process, to make sure that the oxygen content is always well below 5 per cent. Small amounts of oxygen can be removed by passing the gas over palladium pumice, which causes the oxygen to combine in a quiet way with an equivalent quantity of hydrogen.

19 The danger of admixture with air or oxygen is not in any way restricted to hydrogen prepared by the electrolytic process. Air may leak into the suction pipe leading to the compressor, whatever the source of the hydrogen; and this, in fact, is

one of the hazards that must be watched most carefully in handling this gas. The suction line may be fitted with test cocks or drainage valves, and when this is the case it is exceedingly important to keep them closed except when they are opened for legitimate purposes and at proper times. It is believed, moreover, that all such cocks and valves should be kept locked, and that the keys should be in the keeping of some designated, responsible man. To guard further against trouble from this source and from accidental leakage, the whole operation should be conducted so that there will be a positive pressure in the suction line *at all times*. Care should also be taken that the gasometer does not stick and that its seams are kept absolutely tight.

20 It is important to consider the possibility of leakage outward as well as inward. Hydrogen is usually compressed to a pressure of 1800 or 2000 lb. per sq. in., and at high pressures a good deal of gas will escape through a small hole in a short time. If hydrogen gets out into the room and is allowed to collect there, a serious explosion is likely to result. Every discoverable source of ignition should be carefully considered and safeguarded. Rigorous measures should be taken to prevent the men from smoking or using matches or other sources of open flame. The artificial lighting should be provided by incandescent electric lamps enclosed in vapor-proof globes, and it is safest to mount these lamps outside of the windows, locating them so that they shine through the window glass into the interior. All electric wires should be run in closed conduits, and the fuses should invariably be outside of the region of possible danger. This applies also to switches, unless they are of a type which cannot produce an arc or flame. No electric motors should be used in rooms where hydrogen, propane, or any other combustible gas may escape in quantity, unless these motors are specially designed for this particular application, or else thoroughly ventilated by fresh, pure air from the outside of the building. And these various precautions should be observed not only where the hydrogen is compressed, but also wherever it is stored — either in gasometers or in cylinders. Finally, it is exceedingly important to provide abundant ventilation at all times to lessen the chance of fire or explosion in case the various other precautions prove inadequate.

21 Left-handed threads are extensively used for fittings that are to be employed in connection with hydrogen, the purpose being

to prevent the accidental connection of hydrogen lines or cylinders with pipe lines conveying other gases, and especially with those containing air or oxygen. It is to be regretted that in many plants the purpose of the left-handed thread is deliberately defeated by making use of devices known as "adapters." These are in effect right-and-left couplings, which are provided for the express purpose of making it possible to attach a cylinder with a left-handed thread to a pipe line having a right-handed thread.

ACETYLENE

22. No person should ever undertake to compress acetylene unless he fully understands the properties of the gas and knows exactly what it is permissible to do and what he must carefully avoid doing. This is not meant to imply that the compression of acetylene is a dangerous operation when it is performed under the direction of a properly qualified man, but is intended as a special note of warning to those who are tempted to compress the gas without first understanding its properties. Acetylene is an endothermic compound, and this formidable word has frightened a great many persons who do not understand its exact significance. It merely means that heat is absorbed (instead of emitted) when acetylene is formed from its constituent elements. Now it is well known that substances of this kind are likely to be more or less unstable under certain conditions, and acetylene is no exception to the rule; but the conditions under which the instability exists are well known, and it is only the man who does not understand them, or who is pleased to ignore them, who gets into trouble.

23 In compressing acetylene it is important to keep the temperature of the gas as low as practicable at all times. Expressed in more definite language, this means that the compression must be effected by a pump moving with a very low piston speed, and that it must be performed in not less than three stages. It also means that the pump cylinders should be kept as cool as practicable by a plentiful supply of water drawn from the coolest available source. The author cannot state any definite temperature at which the gas may become instable during the act of compression, and in fact it is doubtful if anyone knows what that temperature might be. The only thing that we can say is, that if the operation is conducted as here described, there is apparently no danger from

instability. It certainly is not safe, however, to *store* acetylene in otherwise empty cylinders, at the pressure at which the compressor pumps deliver it. There appears to be a time element involved here, and it is probable that a certain amount of polymerization gradually occurs in strongly compressed acetylene (except when it is stored in special receptacles containing suitable porous material charged with liquid acetone), so that in the course of time small quantities of other more sensitive substances are formed, which serve to lessen the stability. At all events, we know that acetylene should not be stored in an otherwise empty tank at a pressure of more than 15 lb. per sq. in. greater than the pressure of the atmosphere, and yet we know that explosions in the compression pumps, and in the delivery pipes leading from them, are exceedingly rare if the compression is effected as here described. In fact, the author knows of only one case of the sort, and this was precipitated by the breakage of a steel valve spring, which quite possibly struck a spark and thereby caused the development of a high temperature within the tiny space filled by the spark itself.

24 Oil may be used for lubricating the cylinders of acetylene-compressing pumps, but it should afterward be removed by passing the compressed gas through suitable separators as it leaves the compressor. And immediately upon leaving the oil separator the acetylene should be passed into the storage cylinders, which, as previously stated, must be filled with a special porous, solid material thoroughly impregnated by liquid acetone, which dissolves the acetylene and holds it safely in solution.

25 It should not be necessary to issue a warning against mixing other gases with acetylene in the storage tanks, or using acetylene storage cylinders for any purpose whatsoever, except for holding pure, unmixed acetylene (in addition to the safety filling).

ETHYL CHLORIDE, METHYL CHLORIDE, AND PROPANE

26 These gases are used to some extent as refrigerating agents, but it does not appear desirable to discuss their compression in any detail. They are all inflammable, and hence must be handled with due regard to the possibility of fire and explosion. Propane (like butane) does not require lubrication in the compressor cylinder, because the substance itself affords sufficient lubrication. It is necessary, however, to lubricate the stuffing box of the com-

pressor. There has been some difficulty in connection with the lubrication of cylinders in which the chlorides of ethyl and methyl are compressed. Castor oil has been used, but it is not very satisfactory. Glycerine is said to be much better, though still far from ideal. It is said that a new lubricant, consisting of ethylene and propylene glycol mixed with deflocculated graphite, and specially adapted to ethyl and methyl chlorides, and to propane, is now available.

27 In handling these and all other inflammable gases the various special precautions mentioned in connection with hydrogen should also be observed.

AMMONIA

28 Liquefied ammonia is used in immense quantities in connection with refrigerating machinery, and although it perhaps cannot be called a poisonous gas in the narrow sense of the term, it is easily capable of producing unconsciousness and death when it is present in the air in any considerable amount. Similar remarks apply to sulphur dioxide gas, which is also used in refrigerating machinery as well as in various other applications. Both of these gases are freely soluble in water, and it is recommended that in all rooms in which they are handled an overhead sprinkling system be provided that is capable of discharging large volumes of water into the room in case of the accidental liberation of excessive quantities of either gas. A considerable part of the liberated gas would be absorbed by the down-rushing water, and even though the air might not be rendered respirable in this way, something would surely be gained in the way of checking the spread of the gas to other rooms. It is certainly important to provide free and abundant ventilation and adequate means for quick exit in connection with any room in which either gas may suddenly be liberated in quantity. These simple and evident precautions are often wholly disregarded in laying out plants in which ammonia or sulphur dioxide are handled or used.

29 Ammonia gas is usually pronounced incombustible and incapable of being exploded when mixed with atmospheric air, but research and experience have shown that this is not altogether true. A mixture of ammonia gas and air can be exploded if it contains from 16 to 27 per cent of ammonia, and some of the explosions that have occurred in our big refrigerating plants subsequent to the

liberation of considerable quantities of ammonia may possibly have been due to the ignition of mixtures of this kind by means of the arc lamps which have in most cases been in use in rooms where these accidents have occurred. The limits given, while not guaranteed to be the exact ones within which the explosibility exists, are at least approximately correct.

30 In compressing ammonia another source of danger must be carefully considered. If the ammonia gas that is undergoing compression is allowed to become unduly hot, the ammonia that comes away from the compressor will contain a notable quantity of combustible gas that is distinctly different from ammonia. This may come in some measure from the decomposition of the oil that is used in lubricating the compressor cylinder, but it also contains free hydrogen, which appears to be produced by the actual dissociation (or breaking up) of the ammonia into its component gases, nitrogen and hydrogen. We used to think of ammonia as being exceedingly stable and incapable of dissociating in this way to any sensible extent, but we now know that such is not the case. Dissociation occurs even at moderate temperatures, and it is likely to become quite significant when the temperature is high. To guard against the development of combustible decomposition products from the lubricating oil, it is important to use a special kind of oil which experience has proved to be well adapted to work of this kind; and to keep the dissociation of the ammonia itself within as low a limit as practicable, it is important to keep the gas as cool as possible by methods already outlined in connection with other gases. Finally, to guard against vapor explosions in case ammonia gas should escape into the air in considerable quantity, it is important to take the same precautions against ignition sources that have been suggested in connection with hydrogen.

CHLORINE

31 Liquid chlorine is now shipped in vast quantities, for use in connection with bleaching, and for many other purposes. It has been greatly feared by the general public ever since it was used as a military gas in the war, but in time of peace this fear is hardly justifiable. Chlorine is exceedingly irritating, and it can produce unconsciousness and even death if inhaled in any considerable quantity. It has been well described as an "honest gas," however,

the phrase meaning that it has no treacherous qualities. We always know exactly what it will do. When the air on a battlefield was heavily charged with chlorine, there was no place to go in order to be rid of the gas; but in a manufacturing plant, or in a plant in which chlorine is compressed, immediate relief can be had, in case of leakage, by merely leaning out of an open window or going out of the room. It is not desired to underrate the dangers associated with handling liquid chlorine, yet it is only fair to say that they have been largely exaggerated. The gas is far from being like St. George's dragon, and it even has some redeeming features. If a cylinder or a tank containing liquid chlorine bursts or springs a leak, for example, the evaporation that takes place rapidly chills the liquid still remaining in the tank, so that the evolution of gas is quickly and automatically checked, though not wholly stopped. Abundant ventilation is important in a chlorine plant and special attention should be given to the exits, so that if the gas becomes accidentally liberated in considerable quantity the men can pass out into the open air as quickly and directly as possible.

32 Chlorine must be carefully dried before compression, because if it is moist it will corrode the pump, piping, and tanks. Perfectly dry chlorine, on the other hand, has practically no action upon iron. The drying is effected by passing the gas through towers filled with pumice wet with concentrated sulphuric acid. In liquefying the gas by means of a compression pump the main difficulty to be overcome is keeping the piston gas tight in the compression cylinder. A special form of packing is required for this purpose, and it should be designed and installed by some person who understands the necessities thoroughly. In one compression plant visited by the author there are seven sealing rings on the piston — six soft and hard packings of special chlorine-resisting material being used, while the middle space is filled with strong sulphuric acid which not only completes the seal but also serves to lubricate the cylinder. The pressure required in chlorine tanks is not high, as the vapor pressure of liquid chlorine is only about 120 lb. per sq. in., at 70 deg. fahr.; and even when a chlorine tank has been left standing in the sun for a considerable time, the pressure will hardly ever creep up as high as 160 lb. per sq. in. In liquefying this gas the compression is therefore performed in one operation. Tanks and other apparatus used in handling or storing

chlorine must be clean and free from all other substances — solid, liquid, or gaseous — with which the chlorine might combine.

33 In conclusion, there is one general suggestion relating to safety-valves which applies to all gases. Some engineers discourage the use of safety valves on gas-compressing apparatus, partly because the escape of gas through them is a source of waste, and partly because it creates a special hazard, when the gases are toxic or inflammable. But it is not necessary to liberate the escaping gas at or near the work place, and it can often be discharged where it will not constitute a hazard to property. Hence this objection is not altogether well founded. In fact, relief valves should be provided in all cases, and the discharge problem should be considered on its own merits in each individual plant. It is often, and perhaps usually, practicable to have the discharge delivered back into the piping system on the low-pressure side of the compressor, so that there is no economic loss, and no hazard to the men nor to property.

DISCUSSION

Frank Creelman. Acetylene, unlike the other gases considered in this paper, can be exploded without the addition of air or any other gas. This is caused when the temperature is raised sufficiently to cause it to decompose into its constituents, hydrogen and carbon. This decomposition is accompanied by a great evolution of heat which expands the hydrogen, thus causing an increase of pressure.

If the initial pressure of the acetylene is less than 30 lb. gage, and the gas is ignited by heating a spot in the metal container red hot or by introducing an electric spark into the gas inside of the container, only a small amount of the gas, that close to the ignition point, will decompose, and the heat evolved and the pressure caused by this heat will be relatively small, while if the initial pressure is much above the 30 lb. gage pressure the ignition will be propagated through the entire quantity of gas practically instantaneously with great evolution of heat and rapid and great rise of pressure; in fact it detonates.

Dissolving acetylene in acetone does not prevent the detonation of high-pressure acetylene, but if the container is entirely filled

by a porous solid, the pores only containing the dissolved acetylene, the decomposition stated as above will be only local, that is, will not be propagated throughout the mass and no harm will result.

Care, however, must be taken to see that no large voids occur in the container, due either to carelessness in the original placing of the porous material in the container, or to its settling together later, thus leaving voids. If these voids are present the entire gas contents can be detonated.

The real advantage of dissolving acetylene in acetone is that with the proportions of porous material and acetone which is usually placed in the containers, a container holding the gas at any given pressure (say 150 lb. per sq. in.) will hold approximately ten times the amount of acetylene that the same container would hold of the gas (at the same pressure) alone without any porous material or acetone.

There is no evidence whatever that polymerization (cracking) takes place in stored acetylene at ordinary temperatures, even for long-time storage; all experience is to the contrary.

The writer has had many years' experience with compressed acetylene, and would like to say there is no evidence whatever of any such action; in fact, all of our experience is to the contrary, except possibly when the gas compound is compressed to such a high pressure as to be near the critical point of liquification. But no commercial concern today, with the knowledge they have, will attempt to compress gas up to any such point. The gas under these conditions is so very dangerous as to be not fit for any commercial use whatever.

Dr. Risteen also states that the storage of acetylene by this system is the only safe method of keeping it for long periods of time. That does not agree with practice. The dissolved acetylene system is not the only safe system of storing acetylene under pressure. Mr. Max Toltz devised a very safe system which was used extensively by two of our largest railway companies, for many years, with perfect safety from explosions. As a matter of fact, the writer's experience is that it is a little safer than the other.

Dr. Risteen says in compressing acetylene the speed of the pump should be very slow. That is not entirely necessary. In compressing acetylene the speed of the pump has no effect on the gas except in so far as it increases the temperature of the gas. This gas temperature is the limiting feature, so that if the gas is com-

pressed in a sufficient number of efficiently cooled stages, with inter-coolers, also an after cooler (this last is very important) so that the gas temperature does not rise in the hottest spot above, say, 300 deg. fahr., the compressor can be run safely at any speed consistent with these requirements. A sufficient number of good thermometers should be located in the gas passages of the compressor at suitable points to enable the operator to keep track of the temperatures, so that he can shut down promptly before dangerous temperatures are reached.

The compressor should never be allowed to run unless the operator is in the compressor room.

Mayo D. Hersey. May I call attention to the fact that the Bureau of Mines is undertaking a series of investigations on oxygen explosions? That is being done at Pittsburgh. We are to make experiments to show the limiting temperatures and pressures, kind of oil, etc., which determine these explosions, quantitatively.

For our guidance, we shall be glad to be put in communication with all persons having experience with compressed oxygen explosions, particularly where lubricating oil was present, and it may be added that our survey, thus far, strongly corroborates the importance of the precautions suggested by Dr. Risteen.

E. P. Warner. The writer was particularly interested in what Dr. Risteen has said about the use of tetraethyl lead as a means of retarding detonation in acetylene tanks, because of its connection with the use of the same material in a very different field. The General Motors Research Corporation, after several years of work, has added tetraethyl lead as the latest stage in the development of an anti-knock compound for internal combustion engines. The addition of one cubic centimeter of this chemical to every gallon of gasolene is a practically absolute preventative of knock, even in high-compression engines under very unfavorable conditions, as was demonstrated by Mr. Midgley at last summer's meeting of the Society of Automotive Engineers. There is no doubt that the action in the retarding of the production of flame in the burning gasolene vapor mixture in the cylinder is exactly comparable to that in retarding the explosion in the acetylene cylinder.

No. 1862

A NEW SYSTEM OF HELICAL INVOLUTE GEARING FOR USE ON METAL PLANERS

By Forrest E. Cardullo, Cincinnati, Ohio
Member of the Society

In this paper the author gives particulars of a system of helical gearing designed by him for use on metal planers, in which the directions of rotation and the helical angles of the several gears are so chosen as to counterbalance and minimize end thrust, advantage being taken at the same time of the end thrust to counterbalance the side thrust of the cutting tools. The tooth form adopted is an involute having a 14½-deg. pressure angle, a pinion addendum of 3/2 p in. and a pinion dedendum of 1/2 p in., p being of normal diametral pitch. The advantages of this form of tooth for the work in question are enumerated at length and dimensions of a pinion, gear and rack designed according to the principles set forth in the paper are given in tabular form.

THIS PAPER presents the engineering features of a system of helical involute gearing of special tooth form developed by The G. A. Gray Company, of Cincinnati, Ohio, to meet the peculiar conditions necessitated by metal-planer service. It is necessary that planer gears should have the following characteristics in order to give satisfactory service:

a The gears must give smooth and uniform motion without impact, vibration, or chatter. If they do not do so, variations in the driving force, or in the speed of driving, will be transmitted to the table, causing it to vibrate and produce chatter marks on the work

b When the gears wear they must preserve their correct tooth form so that they will continue to give smooth and uniform motion without chatter

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c It is necessary that the gearing have ample strength so that it will transmit the maximum force which can be applied by the source of power, without reaching the elastic limit of the tooth material

d The several gears must have such tooth forms and widths of face that they will run for a reasonable length of time without serious wear

e Should there be wear of the bearings and shafts which would permit the gear centers to separate by a measurable amount, the tooth forms must be such that the gears will still give smooth and uniform motion

f The tooth forms must be such that the teeth can be correctly produced by an efficient and economical machining process.

2 About two years ago, it became the duty of the author to design a system of planer gearing embodying the above requirements. On account of requirements *a* and *b*, it was decided to use a complete train of helical or herringbone gearing, and on account of requirements *e* and *f*, to use the involute tooth form.

3 A study of a design using herringbone gearing showed that within the available limits of space this gearing would not have sufficient width of face to meet satisfactory requirements *c* and *d*. The herringbone gear necessitates either that right- and left-hand gears and pinions shall be machined separately and then fastened together in some way, or else that a sufficient space be left between the right- and left-hand teeth to permit the cutters to run out. The first expedient is not only expensive but it gives a weak and unsatisfactory construction of the pinions. Furthermore, the design of a herringbone rack on this plan offers great difficulties. The second expedient necessitates narrowing the faces of the gears too much.

DISADVANTAGES OF HERRINGBONE GEARING FOR PLANERS

4 A further objection to herringbone gearing for planers is the great practical difficulty of obtaining two helical gears of opposite hand but of exactly equal helical angle, pitch diameter, and pressure angle. A variation in any of these quantities in the two halves of either the gear or the pinion results in an unsatisfactory and "jumpy" action.

5 Helical gearing having all the desirable qualities of herring-bone gearing except freedom from end thrust, it occurred to the author that the hands and helical angles of the several gears might be so arranged as to counterbalance and minimize the end thrust, and that advantage could be taken of the end thrust to counteract the side thrust of the cutting tools. It is usual in planer work to arrange the tools so that they will feed away from the operating position, commonly called the right-hand side of the planer. Also, when a side head is used, it is most convenient and customary to use the right-hand side head. Consequently both the feeding pressure of the rail-head tools and the surfacing pressure of the side-head tool are usually such as to push the table away from the operating side of the machine. These conditions inhere at least 90 per cent of the time in planer work.

6 It is obvious that if the bull gear be made with a right-hand helix, its rotation during the cut will tend to draw the table toward the operating side of the machine and counteract the feeding pressure of the tools. Since this feeding pressure amounts, with usual forms of tools, to about one-tenth of the cutting pressure, the helix angle of 5 deg. 40 min. was fixed upon as being a suitable helical angle for the bull gear and pinion, the end thrust produced by such a gear being approximately one-tenth of the driving force.

7 The remaining gears of the train were made with helical angles of about 12 deg., the hands of the helices being so arranged that the end thrust transmitted from the bull gear to its pinion is distributed between the bull-pinion, intermediate, and pulley shafts, being greater in the case of the slow-moving shafts than in the case of the pulley shaft. Bronze thrust washers, provided with oil by a system of forced lubrication, absorb these end thrusts.

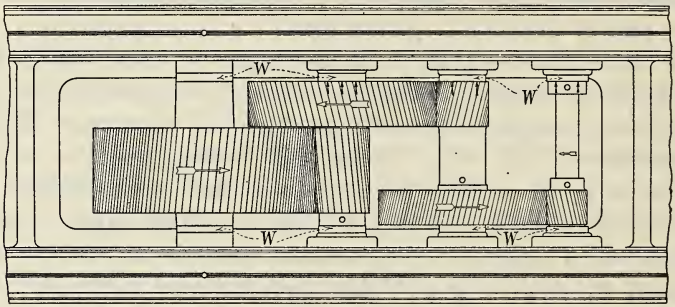
END THRUST OF HELICAL GEARING NOT OBJECTIONABLE IN PLANERS

8 While there may be objections in certain types of machinery to the end thrust produced by helical gearing, these objections do not apply in the case of planers since the driving pressures and its associate end thrusts are relieved each time that the planer reverses, thus permitting complete reëstablishment of the oil film. That the end thrust in this design is not serious may be inferred from the fact that it is only about one-thirtieth of the

end thrust occurring at the end of the worm shaft in the Sellers type of planer drive which the Gray Company have been building for over 30 years and which does not give trouble when working under thrust-bearing pressures of nearly one ton per square inch.

9 The arrangement and general proportions of the train are shown diagrammatically in Fig. 1. The feathered arrows show the direction of motion of the tops of the gears during the cutting stroke, while the arrow heads show the pressure exerted by the gears upon the thrust washers.

10 Imperfections in the action of planer gearing produce a wavy surface on the work known as "gear chatter." If the extreme depth of these waves from crest to trough is not more than



W = Thrust Washer

Fig. 1 Arrangement and General Proportions of the Gear Train Devised by the Author for Planer Operation

0.0001 in. the planer work will be flatter than can be produced by hand scraping, although the waves can be seen quite plainly under suitable oblique illumination. With improperly designed, poorly cut, or worn gearing, the familiar "washboard finish" appears. The waves may be from 0.0005 in. to 0.0015 in. deep, and the work will have to be hand-scraped to a flat surface if a good job is required. The greater the depth of the waves, the greater the amount of metal which must be scraped away. The cost of this work is properly chargeable against the gearing which makes it necessary.

THE FORM OF INVOLUTE TOOTH ADOPTED

11 Consequently no portion of our gear design has received more careful attention than the tooth form. The form adopted

is a $14\frac{1}{2}$-deg.-pressure angle involute tooth, with a pinion addendum of $3/2p$ in., and a pinion dedendum of $1/2p$ in., where p is the normal diametral pitch. In considering the reasons for adopting this tooth form, a brief glance at the theory of helical gearing will be helpful.

12 The only kinematic dimensions which an involute helical gear has are the diameter of the base circle, the number of teeth, and the lead of the tooth helix. Two involute gears whose base-circle diameters, numbers of teeth, and lead of tooth helix are proportionate, will run properly together provided the length and thickness of the teeth are such that they do not interfere. In order to make clear the forms of the tooth faces, consider the cylinders in Fig. 2 with axes AA' and BB' representing the base

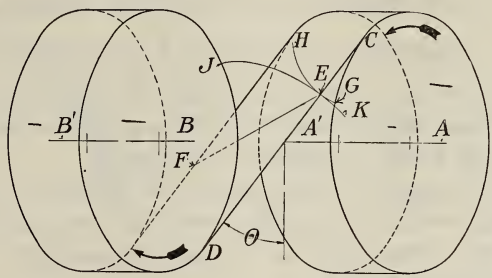


Fig. 2 Development of Involute Tooth Surfaces

cylinders of a pair of helical gears. Joining these cylinders is a flexible inextensible belt, of infinitesimal thickness, and of a width equal to the width of face of the gears. As the cylinders revolve in the direction of the arrows with equal and uniform peripheral speed, the belt unwraps from one and wraps about the other. Any point on the moving belt as point E generates two involute curves as GEH and JEK, one relative to the cylinder from which it unwraps, and the other relative to the cylinder to which it moves. A straight line drawn on this belt *parallel* to the axes of the cylinders will generate in this manner the conjugate active surfaces of two mating *spur* teeth. A straight line such as EF drawn on this belt *obliquely* to the axes will generate relative to the revolving cylinders two warped helicoids which are the conjugate active surfaces of two mating *helical* teeth. Every transverse section of one of these surfaces will be an involute curve. When the belt is wrapped about either cylinder, the oblique line EF

becomes a portion of a helix whose lead is equal to the lead of the teeth. When the belt is moving from one cylinder to the other, the line *EF* becomes straight. Any portion of it lying in the region common to the two gears is a line of contact between the two helicoids which it generates and which form active tooth surfaces. Since at every instant more than one pair of teeth are in contact, there will be two or more limited sections of such straight lines which are the lines of contact between different pairs of teeth. The aggregate projected length of these lines of contact upon the axis of either cylinder is called in this paper the *effective lineal contact.* The pressure at the lines of contact in pounds per inch of effective lineal contact is called the *specific contact pressure.*

ADVANTAGES OF THE TOOTH FORM ADOPTED

13 The advantages of the tooth form adopted are:

a The use of a short dedendum in the pinion teeth eliminates interference, which is a prolific source of trouble in all kinds of machine-tool gearing.

b The strength of the pinions is increased about 40 per cent without using stub teeth or increasing the pressure angle. This may be seen by comparing the tooth forms shown in Fig. 3

c The use of long addendum and short dedendum for the driving pinions shortens the angle of approach and lengthens the angle of recess. This type of tooth action is sometimes called "long follow through" and gives great smoothness of motion for the following reasons:

1 During the angle of approach the effect of tooth friction in causing vibration is cumulative, the friction increasing the effective pressure angle and the tooth pressure, which in turn increases the friction. This exaggerates the periodic variation in the amount and direction of the tooth pressures. During the angle of recess the increase in friction reduces the effective pressure angle and the tooth pressure, thus damping any vibrations produced by variations in friction. The effects of these actions are similar in a general way to those obtained by pushing a flexible pole along a rough road in front of one in

the case of the angle of approach, and dragging it after one in the case of the angle of recess. In the one case the vibration is very marked. In the other case it is scarcely noticeable

2 The active portion of the tooth profile is removed from the base circle so that the involute curvature changes gradually, giving a smoother action

3 During the angle of recess there is less relative sliding than during the angle of approach, which reduces tooth wear.

d The diameter of the pinions at the bottom of the teeth is increased, giving stronger pinion bodies

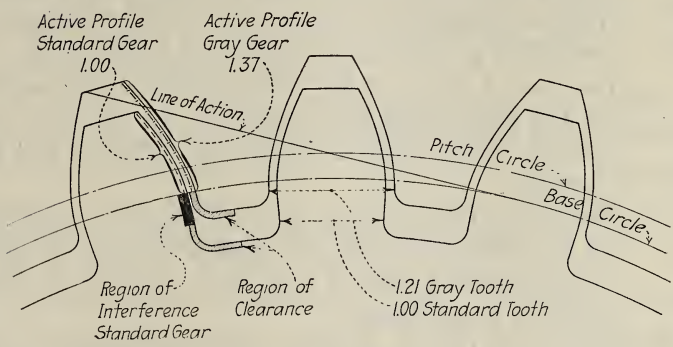


FIG. 3 COMPARISON OF THE GRAY AND STANDARD FORMS OF INVOLUTE TEETH

e The short dedendum avoids undercut or radial flanks with their attendant evils.

14 In order that the advantages of long-addendum drivers shall not be lost in the case of the bull gear working against the rack, the bull pinion, bull gear, and rack are of peculiar kinematic properties. The bull pinion is a long-addendum driver of $14\frac{1}{2}$ deg. pressure angle, and the bull gear, when meshing with the pinion, is a long-dedendum driven gear of the same pressure angle and cut with an exactly similar hob of opposite hand. When the bull gear meshes with the rack, however, it becomes a long-addendum driver, the change being produced by appropriately reducing the pressure angle, the circular pitch, and the helical angle of the rack. The bull gear accordingly has two pitch circles, the larger one associated with the bull pinion and the smaller one with the rack.

Each pitch circle has its own circular pitch and its own helical and pressure angles. Were it not for this design, most of the action of the bull gear and the rack would occur during the angle of approach, when the circumstances are unfavorable to smooth action. By this peculiar construction, however, we achieve the following advantages:

a The bull gear drives the rack mostly during the angle of recess, giving the advantages already described in connection with the long-addendum pinion

b The pressure angle of the bull gear against the rack is theoretically only about 8½ deg., and is practically reduced still lower by the tooth friction, greatly diminishing the tendency to vertical vibration[1] arising from imperfect gear action or variation in tooth friction

c The line of action of the force applied by the bull gear to the table being more nearly in the direction of the motion of the table, is much more effective and satisfactory

d Because of this very low pressure angle the tendency of of the bull gear to lift the table is only about half the usual amount,[2] making hold-down gibs and side-thrust bearings unnecessary

e Because of the low pressure angle the number of teeth in contact is greatly increased, which confers the following advantages:

[1] Gear chatter is caused by periodic variations in the amount and direction of the tooth pressure. Such variations when finally transmitted to the table rack have two components, one in the direction of motion and the other in a vertical direction. The first of these components is harmless, the second produces vertical movements of the table and work. A low pressure angle between the bull gear and table rack tends to eliminate this vertical vibration.

[2] In this connection the author desires to point out that conditions occasionally arise when a left-hand side head is being used or heavy roughing is being done by feeding the tools toward the operating side, that the end thrust of the bull gear adds to instead of counteracting the tool pressure. Under these unusual conditions the bull gear pushes the table against the right-hand surfaces of the ways. This tends to cause the table to lift. However, the pressure angle of the bull gear in the Gray design is such that even under these unusual and unfavorable conditions the tendency of the table to lift is less than the tendency of a spur-geared table to lift under the most favorable circumstances.

1 The load per tooth is reduced in much greater proportion than is the strength of the tooth, so that the construction is actually stronger than would be the case with the larger pressure angle

2 The effective lineal contact is greatly increased, reducing the specific contact pressure and consequent wear.

REASONS FOR EMPLOYING A FULL-LENGTH TOOTH AND A LOW PRESSURE ANGLE

15 A few words of discussion as to the reason for choosing the full-length tooth, that is, a tooth where the working depth is equal to 2 divided by the diametral pitch, and also for choosing the low pressure angle, are in order.

16 In the case of spur gears which have become slightly worn, some wear occurs at the tips of the teeth and still more on the flanks of the pinion where the tips of the gear teeth rub. Theoretically a spur gear gives tooth contact at all points lying on the line of action, and during part of the time more than one pair of teeth may be in contact. Actually, as soon as the teeth wear, only one pair of teeth will be in contact at a time. Therefore the shorter these teeth, the stronger they will be. In the case of helical gearing, because of pitch-line rolling, the wear is distributed equally over the entire surface of the tooth and the theoretical number of teeth always remain in contact, the effect of the wear being solely to rectify any errors in cutting and increase the backlash. The strength of helical gearing is therefore not appreciably increased by using stub teeth, since lengthening the teeth increases the effective lineal contact, which is equivalent to increasing the width of face of a spur gear. On the other hand, the greater number of teeth in contact and the longer arc of action give much smoother tooth action. Furthermore, the lower specific pressures increase the efficiency, have less tendency to break down the oil film, and greatly reduce the wear.

17 The 20-deg. pressure angle seems to have certain theoretical advantages over the form adopted which do not bear the test of experience. The higher pressure angle eliminates interference, reduces relative sliding and consequent friction, and gives a stronger form of tooth. However, these same advantages inhere in the

tooth form adopted, and it is found that the lower-pressure-angle gears give smoother and more satisfactory action. The reason for this is threefold: In the first place, variations in friction produce much less marked variations in the direction and amount of the forces between the teeth. In the second place, the lower pressure angle reduces the objectionable vertical component of the vibration. In the third place, the low pressure angle gives a longer arc of action, more teeth in contact, and a greater effective lineal contact.

TABLE 1 KINEMATIC DIMENSIONS OF A PLANER BULL PINION, BULL GEAR AND RACK DESIGNED ACCORDING TO THE AUTHOR'S SYSTEM OF HELICAL INVOLUTE GEARING

	Pinion	Gear with Pinion	Gear with Rack	Rack
No. of teeth	20	80	80	∞
Pressure angle (transverse)	14° 34′	14° 34′	6° 51′	6° 51′
Base-circle diam., in.	4.862	19.45	19.45	∞
Helix angle (at pitch line)	5° 40′	5° 40′	5° 32	5° 32′
Pitch diameter, in.	5.024	20.097	19.597	∞
Circular pitch (transverse), in.	0.78926	0.78926	0.76962	0.76962
Addendum, in.	0.375	0.125	0.375	0.125
Dedendum, in.	0.125	0.375	0.125	0.375
Angle of approach	10.9°	.....	6.35°	.....
Angle of recess	21.5°	.....	10.6°	.....
No. of teeth in action	1.8	1.8	3.75	3.75
Actual width of face, in.	8½	8½	8½	8½
Effective lineal contact, in.	15.3	15.3	32	32
Hand of helix	left	right	right	left

The actual advantages were found to lie with the low pressure angle.

18 It is obviously desirable that the faces of the gears should be as wide as possible in order to increase the strength of the gearing and reduce the specific contact pressure. Furthermore, with the low helical angle chosen for this design, wide-faced gears are necessary to secure overlapping tooth action. In order to permit of gears of maximum width of face, the rack is offset so that its center line lies to one side of the center line of the table. This permits the rack to be widened about 40 per cent and also the intermediate gear to be widened. The center line of the rack is placed toward the right-hand side of the planer, since the right-hand side head is the one generally used and since narrow work placed on the table is generally placed close to the operating side of the

machine. In this manner the line of action of the driving force is brought nearer to the line of action of the cutting tools, reducing the tendency of the tool pressure to turn the table in a clockwise direction when viewed from above. Incidentally, the vertical component of the bull-wheel pressure tends under the usual operating conditions to counteract the downward component of the tool pressure, which would otherwise bear more heavily on the right-hand V.

19 Table 1, which gives the kinematic dimensions of a bull pinion, bull gear, and rack designed in accordance with the foregoing principles, will be of interest.

DISCUSSION

Charles Meier. The author's contentions in Par. 1 are correct as far as they go; however, it would be interesting to learn something of the efficiency of this system of gearing which he describes, as it has always been an outstanding feature in the design of planer transmission, as in all other designs of heavy machinery, that the efficiency of the gearing should be as high as possibly obtainable, owing to the high power factors which must be transmitted and their correspondingly high speeds.

It seems to the writer that the gearing as described by the author involves more friction losses due to the great amount of sliding motion on the gear teeth which is bound to lessen correspondingly the efficiency when compared to the standard $14\frac{1}{2}$-deg. involute tooth system in vogue today in which the principal aim is to eliminate, as far as possible, all sliding motion and produce a truly rolling action, which eliminates undue wear and increases efficiency. It seems there would be an appreciable amount of end thrust which would necessitate using thrust washers at the ends of the shafts. This is shown by the statement that during the reverse motion the washers are relieved so as to admit a filament of oil. On the cutting stroke the reverse must be true, and it seems that the heavier the cut the greater will be the end thrust and, in the writer's opinion, a corresponding falling off in efficiency.

The author refers to the end thrust as not objectionable, citing the case of the Sellers type of drive, yet it has been found that this type of drive does not lend itself to the present-day high speeds owing to its low efficiency. Also that in most cases an adjustment is provided to compensate for wear due to end thrust.

While it is true that the undercut at the dedendum with the standard 14½-deg. system is objectionable and is usually the cause of uneven motion to the table, this can be eliminated entirely if the pinions will be made to contain 20 teeth or more and with the single purpose system of cutters, if properly made to correspond exactly to each pair of gears, it will be found that these produce an equally smooth system of gearing with as nearly true rolling action as it is possible to obtain with a correspondingly higher efficiency.

It is the writer's opinion that far greater accuracy will be required in mounting this system of gearing than would be required by the ordinary system of straight-faced gearing, as any slight discrepancy in the angularity or the locating of the gears will cause a difference in action of one gear upon the other which is also true if one gear would wear faster than the other. It would seem that any such conditions would cause uneven rotation and a jerky motion to the table.

With reference to the helix angle of the bull wheel offsetting the side thrust of the cutting tools, it has been the writer's experience that at no time is the table in danger of being tipped out of the ways unless the table is too light or too short for the work in hand.

If the table is of proper proportions the side thrust of the tools will readily be accommodated by the ways in the bed and there will be no danger of tipping and the use of holding down gibs can be eliminated.

It will be found that if the table is pushing out of the V's the following action takes place: The tools enter the work, the front end lifts, but as the table moves on and the tools reach the middle of the table it settles back into the V's until a certain distance at the back is reached, where it is again lifted until the tools leave the work. I do not believe it possible to overcome this condition by any system of gear-tooth design.

F. K. Hendrickson. Close inspection and observation of the old standard type of planer drives would without doubt show the chatter marks made by the spur gear drive. For some years the Whitcomb-Blaisdell Company of Worcester has encountered this condition, and in order to overcome the difficulty, has finally resorted to the use of helical gears after other extensive and expensive experiments.

By the introduction of helical gears, the chatter marks have disappeared, as well as the vibration in the machine. Special 14½-deg. gears of angular involute forms were used with most gratifying results. The gears were recommended by a well known authority on gearing. While these gears were probably not exactly of the same form as described in Mr. Cardullo's paper, they resemble them in a great many features. It is questionable, however, if it is absolutely essential to resort to this special form, as described by Mr. Cardullo, as it seems to be the general opinion that the standard involute form would work out satisfactorily in the helical gear drive.

I would like to ask Mr. Cardullo if in his development he has found the standard involute helical gear impracticable and if he has definitely established, by experiment or otherwise, that the elimination of the chatter marks can be overcome only by the use of his new tooth form.

The paper states that the side pressure caused by the feed is one-tenth of the tool pressure of the average planing tools. In his endeavor to overcome this side pressure the author has opposed the angle of the spiral to the feed pressure of the cutting tool. In the Whitcomb-Blaisdell gearing the angle of the tooth has been arranged to force the table away from the operator, which in reality adds to the feed pressure the angular pressure of the helical tooth. The actions of both pressures were purposely made in one direction, so that the special V tracts and troughs would counteract the difficulty. It is, therefore, our opinion that with the tipped V and this arrangement of gearing, that there need be absolutely no fear of table side thrust, and climbing tendency in the V is due to any excessive pressures.

The author also brings out another feature, which is worthy of comment, namely the placing of a rack at the right hand side of the table, more directly below the line of action of the average cutting position. I feel that this is correct, were it not necessary, due to the extra width of gear faces.

The author mentions that gear chatter is caused by periodic variations in the amount and direction of the tooth pressure and that such variations have two components, one in the direction of motion and the other in a vertical direction. He advises that the first or the one in the direction of motion is harmless.

When it is considered that the tools are held in a built-up holder, known as the clapper box, on the cross rail, it is hard to

agree that sudden spurts of speed, which in reality cause momemtary excess pressure on these tools, will not be at all harmful. It is my contention that however short these periodical variations lengthways of the bed may be, the tools will cause a deeper cut into the work causing in a measure part of the so-called chatter marks. Therefore it is not reasonable to assume that all the chatter is caused by the vertical action alone.

Earle Buckingham. We are coming to consider gear design exactly as we consider the design of other machines; that is, where unusual conditions are present, to design the gears to meet them, rather than merely to follow conventional practices. A gear profile, in effect, is nothing more or less than a cam; and there are certain principles that are followed in designing cams, which can be applied with very great advantage in designing gears.

Mr. Meier stated that he thought Mr. Cardullo's design would increase the sliding. If he made an analysis of the action he would find that it does not necessarily increase it, but often reduces it. He also spoke of the effect of small errors in alignment, as being greater with a helical gear than with a spur. The spur gear has a helix angle of zero, and there is an error in the angle if the gear is slightly helical instead of straight, and there will be trouble.

G. M. Eaton. I want to say a few words on helical gears though not applied to planers, but to railway and industrial gears. We began about ten years ago to experiment with helical gears, with what we call a flat angle. There was brought to our gear works a request to make a flat-angle helical gear. We were immediately impressed with the results. I looked after the gear for 60,000 miles of operation on a high-speed, interurban car, and compared it with spur gears, and certain other freak forms that we were experimenting on and found the helical gear absolutely outstanding in its superiority. The efficiency of the gear was tested, and found to be lower than that of the spur gear. The loss was less than 2 per cent, so that it did not amount to anything compared to the advantages of longer life and the smoothness of operation. The tooth form was developed by analytic methods. The strength feature was essential because of the severe service. There are 30,000 of these gears now in service and they are standing up well.

A word about end-thrust; our customers were told that thrust collars would wear out in very short time because, in a railway

application, it would be necessary to run perhaps for an hour with the end-thrust on one collar. Some of these gears were installed on the Chicago, Milwaukee and St. Paul locomotives, where the run is for an hour up the mountains, and having measured all the gears in service by the collars, we find no difference in any thrusts. We examined a 2500-hp. motor that had run up to a year and a half in service, and tabulated the end wear, and found 25 per cent less end wear on the helical gear than on the spur.

The Author. Mr. Meier's question regarding the efficiency of this gearing is correctly answered by Major Buckingham. The tooth form is more efficient than the conventional tooth form. The horsepower lost at the thrust washers, with the type of lubrication employed, will not exceed $\frac{1}{5}$ of one per cent of the power supplied. There is no measurable difference between the efficiency of the best type of splash lubricated spur gearing, and the gearing described.

Mr. Meier is mistaken in thinking that interference disappears with a 20-tooth pinion, in the case of the standard $14\frac{1}{2}$-deg. involute gear. The minimum number of teeth free from interference is 31.

When helical gears are properly cut with equal helical angles they require no greater accuracy of mounting than spur gears. Helical gearing is sometimes carelessly cut so that the helix angles are unequal, and such gears will give trouble, but if the engineering department does its work correctly it is just as easy for the shop to produce correct helical gears as to produce correct spur gears. When mating spur gears wear unequally bad results follow. When mating helical gears wear unequally, continuous pitch line rolling prevents the uneven rotation which Mr. Meier mentions.

Mr. Hendrickson asks whether spur gearing, and standard $14\frac{1}{2}$-deg. teeth, necessarily produce chatter marks. That I cannot tell. From theoretical considerations I would say that if a spur gear were cut absolutely true, and were operated at the proper speeds, and under proper conditions, chatter marks would not occur. Our own experience in the planer business shows that once in a great while a spur gear machine is sensibly free from chatter marks, when new, but soon begins to show these marks.

Two surfaces, made by any type of machine tool driven by gearing, and rubbed together, will show parallel bright lines caused by gear chatter. With the type of gearing described, one may always be sure that the maximum elevation of such chatter marks

is too small to measure with ordinary instruments, and the flatness of a planed surface of homogeneous material is superior to that of a scraped surface. On the other hand, with spur gears one cannot tell which gears will give inferior results until they are tried. The system of gearing proposed in the paper eliminates the uncertainty.

Other tooth forms used for helical gearing, after they wear into form, will give satisfaction. When they are new some of them will, and some of them will not, give satisfaction. It is a question of how they are cut. The point is to get as near theoretical perfection as possible to start with, and to eliminate as far as possible, by the design, those difficulties that proceed from the imperfections of workmanship.

No. 1863

STEAM UTILIZATION IN A MODERN NEWSPRINT MILL

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Associate Members of the Society

The paper is a detailed study of the conditions found in a modern paper mill, with which the authors have been associated, for the purpose of determining the heat required. The mill and the paper machines are described. An analysis is made of the conditions existing in the drying operations and the results of tests on the drier together with the theoretical heat requirements are given. The amount of heat necessary for heating and ventilating the machine room is investigated and figures for the amount of heat involved are given.

THE purpose of this paper is to show the heat required in the manufacture of newsprint on a modern basis. It is, in general, a detailed study of the concrete conditions found to exist in a modern paper mill with which the authors have been associated. Other conditions found to exist in paper mills have been treated, however, with a view of showing their bearing upon the different subjects which are covered herein, and also to facilitate comparison by other mills whose installations are different from the one cited.

2 In general, the importance of heat conservation has not been appreciated in this industry and, in view of the national movement to conserve natural resources, all newsprint paper mills are in duty bound to study their own particular problem, so that their coal or fuel-oil requirements may be reduced to a minimum. It is hoped, therefore, that this paper may be of assistance to those who, no longer content with operating by rule of thumb, desire to analyze their heat load, in order to assist in conservation and to reduce the cost of manufacture.

3 A survey made as of January 1, 1922, shows that the United

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States and Canada have a combined daily output of about 8500 tons. The tonnage produced during the year 1921 was approximately 2,038,095 tons, and the value of this tonnage, at a representative figure of $100 a ton, would be $203,089,500.

4 On the basis of the familiar statement that "One ton of coal is required to produce one ton of paper," the newsprint industry has a yearly fuel demand equivalent to about 2,040,000 tons of coal. In the United States, it has been quoted as occupying sixth place in capital invested and value of product, while in horsepower installed it probably ranks fourth.

DESCRIPTION OF MILL INVESTIGATED

5 The building housing the two paper machines is 288 ft. long by 88 ft. wide and has a total height of 45 ft. It is of standard brick and steel construction, with concrete floors and roof, the latter being waterproofed with tar and felt. The paper machines are carried on an operating floor which is 27 ft. from the roof. Underneath is a basement having a height of 18 ft. and its floor at grade level. Only the northeast side is exposed, other buildings adjoining the remaining side and ends. The openings in the exposed wall have a total area of 3000 sq. ft.

6 The greater part of the driving mechanism for this installation is located in the basement. The prime movers are steam turbines of the single-stage non-condensing type, direct-connected through flexible couplings to reduction gears, and have a rated capacity of 400 b.hp. at a speed of 2675 r.p.m. This gives a speed reduction of about 7 to 1. The turbo-gears in turn drive an English system of rope transmission, in which each section of the paper machine has its own jackshaft, carrying a cone pulley and sheaves.

7 By means of belts, power is transmitted from the cone pulleys on the jackshaft to the cone pulleys driving each section of the paper machine. The cone pulleys transmit through a friction clutch and herringbone gears to each of the sections. In the case of the drier sections further spur gears are used to drive from one drier to the next in the section.

THE PAPER MACHINES

8 The paper machines are of the Fourdrinier type, are 166 in. wide and have three presses. The drier sections consist of one

fore drier, 24 in. in diameter by 162 face; 32 driers, 60 in. in diameter by 162 in. face; and two felt driers, for the first section felts, 48 in. by 162 in. All driers are of cast iron, bored on the inside, and turned and polished on the external surfaces. Each drier is equipped with a dipper, to remove the water of condensation. These machines are designed for a maximum speed of 700 ft. per min.

9 Newsprint consists of from 70 to 85 per cent of ground wood and the remainder sulphite pulp. After the mixture has been reduced to a consistency of about $\frac{1}{2}$ per cent, it is caused to flow onto the wire of the paper machine in amounts carefully regulated so as to finally produce finished paper weighing 32 lb. per ream of 500 sheets, size 24 by 36 in.

10 The web is formed by the oscillating motion of the Fourdrinier section. Water is removed through the wire, assisted by the capillary attraction of the table rolls, which also act as carriers for the wire. The suction boxes remove a large quantity of water and assist in closing the sheet. The paper then passes between the couch rolls, which press out additional moisture.

11 In passing through the press sections the sheet is carried on woolen felts between weighted rolls, further reducing the moisture content.

12 The paper is then ready to enter the driers, where water is removed from the sheet by contact with steam-heated cylinders and is carried through same on canvas felts. The last drier is equipped for cold-water circulation, so as to dampen the sheet before it passes through the calender stack for ironing and polishing, prior to cutting to specified size and winding on cores.

HEAT REQUIRED FOR NEWSPRINT MANUFACTURE

13 In order to show more clearly the heat required for newsprint manufacture, simultaneous tests were conducted on one of the previously mentioned machines to determine the heat demand for driving, drying and ventilating.

14 The data pertaining to the turbine are given in Table 1. The steam supplied to the turbine was the same as that used by the driers. In other words, the turbine water rate balanced the drier heat demand and no auxiliary live steam was required.

TABLE 1 TURBINE TEST

Initial steam:	
Pressure, lb. per sq. in. abs.	160.7
Temperature, deg. fahr.	401.3
Total heat per lb., B.t.u.	1,217.3
Exhaust steam used by driers:	
Pressure, lb. per sq. in. abs.	23.7
Temperature, deg. fahr.	248.1
Total heat per lb., B.t.u.	1,164.7
Lb. steam per hour	16,822.00
Lb. steam per min.	280.37
B.t.u. per min. to turbine	14,747.30
Equivalent B.t.u. per b.hp. (as given by manufacturers)	44.17
B.hp. developed	334.00
Water rate, lb. per hp-hr.	50.37

15 In selecting a suitable prime mover the governing factor is the amount of dry saturated steam required for drying the paper; consequently it does not follow that the engine or turbine having the lowest water rate is the most suitable.

16 The water rate of the prime mover, in conjunction with the initial steam pressure and quality, must also be such as to insure approximately dry exhaust steam, at drier pressure, as it is undesirable to have steam of a high moisture or superheat content entering the driers.

17 Fig. 1 has been prepared to show the effect of the water rate of the turbine upon the quality of the exhaust. It will be noted that for the rate existing at the time of the test the exhaust was superheated 11 deg. fahr., and that when 45 lb. were used per b.hp. per hour the exhaust steam would be dry-saturated.

18 In the past it has been very difficult to obtain data showing the horsepower requirements of paper machines, but with the advent of motor drives more information is becoming available. It is impossible to give a formula that will cover all installations, for the power requirements will vary with the type and mechanical design of the machine, the conditions under which it is operated and the efficiency of transmission. The following, however, is cited to show approximately the power demand of the variable-speed drive, and is based on a speed of 100 ft. per min. and 100 in. width:

1 Modern high-speed motor-driven machines, having four presses, forty 72-in. driers and operating at speeds varying between 700 and 1000 ft. per min. . . . . 25 to 30 hp.

2 Turbine-driven machines, rope-drive transmission, having three presses, thirty-two 60-in. driers, and operating at speeds of 550 to 700 ft. per min. .....30 to 34 hp.

3 Engine-driven machines, Marshall drive, having three presses, thirty-two 48-in. driers, and operating at speeds of from 350 to 600 ft. per min.....33 to 38 hp.

ELECTRIC DRIVES FOR PAPER MACHINES

19 Much attention of late has been given to electric drives for paper machines, and in some recent installations the electric

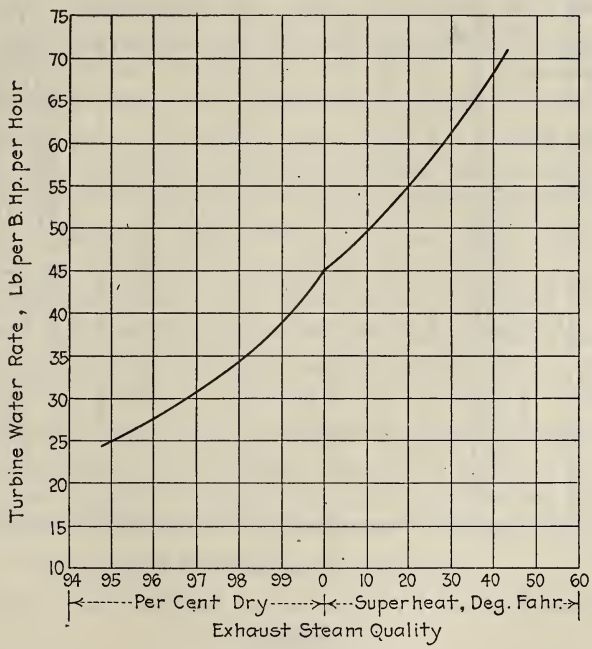

Fig. 1 Curve Showing Effect of Water Rate on Quality of Exhaust

energy for driving is generated by a hydroelectric plant. The steam required for the driers is then obtained from high-pressure steam reduced through a pressure-reducing valve to a lower pressure, with consequent superheat; as will be shown later, however, superheat is not suitable for use in the driers.

20 Further, unless electrical power can be purchased at a cheap rate, it is not thought that such an installation is economical,

and for the purpose of illustrating this point the following computations have been made to determine at what price electrical energy would have to be purchased to be the equivalent of steam assumed to cost $0.75 per 1000 lb.

21 Referring to Fig. 2, which is based on the test given in Table 1, straight line *A* (205.1 B.t.u.) represents the heat of the drier condensate, and line *C* (1217.3 B.t.u.) the total heat of one

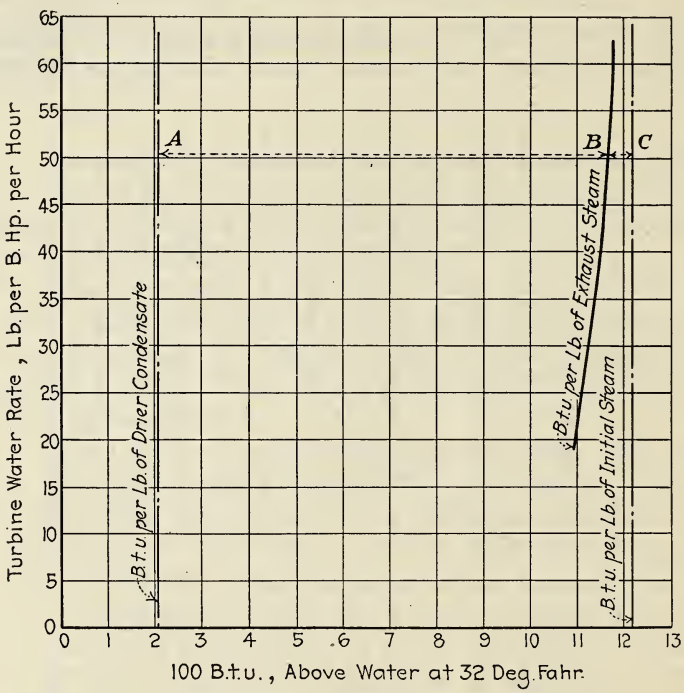

FIG. 2 CURVE SHOWING B.T.U. PER LB. OF EXHAUST STEAM AVAILABLE FOR DRIERS

pound of steam under the conditions given. Curve *B* shows the relation between the water rate of the turbine and the heat contained in the exhaust available for drying.

22 *AC* (1012.2 B.t.u.) therefore represents the heat per pound of steam which would be available for drying if a pressure-reducing valve was used. *BC* (52.6 B.t.u.) represents the amount of heat abstracted from the initial steam in doing work in the turbine.

23 It will be seen that the quantity expressed by *BC* is small in comparison with that represented by *AB* (959.6 B.t.u.), and that the exhaust steam contains nearly as much heat as the initial steam.

24 From the preceding data and based on 7600 operating hours per year, to develop one brake horsepower-year, 20,136,000 B.t.u. are required. This is equivalent to 19,900 lb. of steam which, at an assumed cost of 75 cents per 1000 lb., makes the steam cost of a horsepower-year, approximately $15.

25 Assuming the efficiency of the electric drive to be 85 per cent and that of the drive described to be 75 per cent, the cost of purchased electrical power must not exceed $17 per hp-yr.

26 This comparison does not take into consideration interest and depreciation on capital invested or the maintainance and operation costs; but as the initial cost of the electric drive is approximately two and one-half times that of a turbine rope-drive installation, it is thought, that if these costs were considered, the comparison would be favorable to the turbine rope drive.

27 The electrical installation, however, offers many advantages from the standpoint of operation, and with the development of high-speed machines will no doubt become the most acceptable type of drive. It is the opinion of the authors that the application of the motor drive will prove most economical when the necessary electrical power is developed by a turbo-generator, the exhaust from which would be utilized for drying the paper.

28 The bleeder or extraction type of turbine would have the advantage of greatest flexibility in that a minimum amount of steam would be wasted during periods of light loads, when practically no steam is required in the driers.

DRYING PAPER

29 In the drying of paper it is requisite that the moisture be removed gradually to prevent scorching and cockling of the sheet, and this requires a definite temperature gradient through the machine. It is also necessary that the temperature of the paper be kept within certain limits to produce a good, strong sheet.

30 The authors believe that the logical method to employ in these heat calculations is to base all figures on the pounds of moisture removed from the paper, instead of on the actual production.

This is due to the fact that the moisture content is variable, depending upon the number and efficiency of the machine presses and the amount of moisture desired in the finished sheet.

31 The amount of moisture in the paper entering the drier section generally varies from 65 to 76 per cent, with an average of about 72 per cent. The amount of moisture in the finished sheet varies from 5 to 10 per cent, with an average of about 8 per cent. It is important that enough moisture remain in the finished sheet to insure proper finish and strength.

32 By the use of the following formula the number of pounds of moisture to be removed per pound of finished paper can be readily determined:

$$W = \frac{M_1 - M_0}{100 - M_1}$$

where W = pounds of moisture removed per pound of dried paper
M_1 = percentage of moisture in sheet entering drier section, and
M_0 = percentage of moisture in dried sheet.

33 It is important that as much moisture as possible be removed from the sheet before it enters the driers. It should be understood, however, that there are limitations to the amount of moisture that can be removed by mechanical means, namely, the possibility of crushing the delicate web of newsprint and the effect on the life of the press felts. Each mill, by considering the above in conjunction with the cost of steam, should determine the most efficient point of moisture removal.

34 In order to determine the economical life of machine clothing, in relation to steam cost for drying, a study of results obtained in several mills was made. This showed that the relation between life of felts and the percentage of moisture entering the driers followed a straight line, and that it is the combined felt-days of all the press felts which determines the moisture in the sheet entering the driers. This is a fortunate relation, in that the operating days of each of the three press felts is seldom the same at any one time.

35 Fig. 3 shows the relation between the combined felt-days of the three press felts and the average percentage of moisture

leaving the presses for the period considered; this period being taken as one-third of the combined felt-days.

36 Table 2 is self-explanatory and shows the relation between combined felt-days of the three presses and the cost per day, in dollars, for steam and felts. The calculated costs do not take into account the labor and the loss of production due to changing clothing. Further, this study only applies for a par-

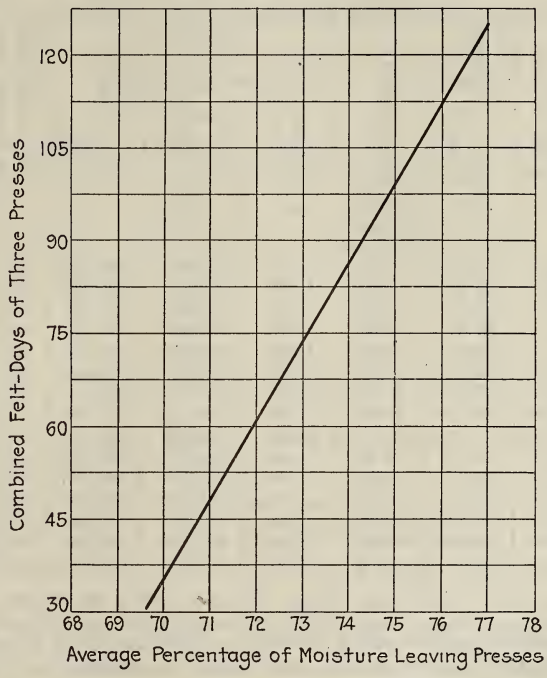

Fig. 3 Curve Showing Life of Felts in Relation to Moisture in Sheet Leaving Presses

ticular period of fixed steam and machine-clothing costs, and should be revised as conditions change.

37 Some plants, thinking to economize, have tried to dispense with drier felts. This procedure has not proved satisfactory, for the reason that intimate contact of the sheet with the drier surface is of vast importance in heat transfer. The drier felt also assists in the removal of moisture.

38 It has been noted that a new drier felt results in a decreased steam consumption of about 300 lb. per ton of production as compared with an old one. This is due to the fact that as the felt ages, the meshes gradually become closed.

39 In investigating the action of the driers, that portion of them in intimate contact with the sheet has been termed the "effective" surface, while the remainder has been called "free"

TABLE 2 COST OF STEAM AND FELTS

Felt-days. (1)	% Moisture entering driers at end of period (2)	Avg. % moisture for period (3)	Period in days (1) ÷ 3 (4)	Moisture evap. per lb. of paper (5)	Lb. steam per day F. & A. 212° F. $\frac{(5)\times5400\times24\times1361}{970.4}$ (6)	Cost of steam per day. Dollars. $\frac{(6)\times0.75}{1000}$ (7)	Cost of press clothing per day. Dollars (8)	Cost of steam and felts per day. Dollars. (7) + (8) (9)
30	69.60	68.40	10.00	1.911	347,360	260.52	68.00	328.52
35	69.95	68.58	11.67	1.928	350,440	262.83	58.20	321.03
40	70.30	68.75	13.33	1.944	353,347	265.01	51.00	316.01
45	70.75	68.98	15.00	1.966	357,347	268.01	45.35	313.36
50	71.10	69.15	16.67	1.982	360,253	270.19	40.80	310.99
55	71.45	69.33	18.33	2.000	363,533	272.65	37.10	309.75
60	71.90	69.55	20.00	2.021	367,347	275.51	34.00	309.51
65	72.25	69.73	21.67	2.039	370,613	277.96	31.40	309.36
70	72.65	69.93	23.33	2.060	374,440	280.83	29.10	309.93
75	73.05	70.13	25.00	2.080	378,067	283.55	27.20	310.75
80	73.45	70.33	26.67	2.101	381,893	286.42	25.50	311.92
90	74.20	70.70	30.00	2.140	388,973	291.73	22.65	314.38
100	75.00	71.10	33.33	2.183	396,800	297.60	20.40	318.00
110	75.80	71.50	36.67	2.228	404,973	303.73	18.55	322.28
120	76.60	71.90	40.00	2.274	413,333	310.00	17.00	327.00

NOTE: 8 per cent moisture in dried paper. Steam cost assumed at $0.75 per 1000 lb. No tests are available for moisture entering driers with complete new press clothing, but from a study of results which might be expected, it is assumed that moisture entering under these conditions would be 67.2 per cent.

surface. In making comparisons between different machines the ratio of the effective surface to the total surface should always be stated.

40 The amount of radiating surface which does no direct useful work increases with the drier diameter, and to show the effect of diameter on effective surface, Table 3 has been prepared for a 166-in. drier with an actual working face of 156 in. This table is of interest in connection with the design of modern machines, where the high speed required must be obtained either

through large diameter of driers or the greater relative speed of those of smaller diameter.

41 To obtain the maximum value of the ratio E/T, the drier section should be designed so as to obtain as large an arc of contact as possible, consistent with ease of handling paper and changing drier felts. Experience shows that with an open pit under the drier section the life of the felts is materially increased, owing to the better ventilation afforded.

42 There are two methods in general use for removing condensation from driers, namely, siphons and revolving dippers; and opinions are at variance as to which is to be preferred. Unfortunately, neither system is capable of completely removing all

TABLE 3 EFFECT OF DRIER DIAMETER ON EFFECTIVE SURFACE

Diam. of driers, ft.	Total surface in sq. ft. (T)	Effective surface in sq. ft. (E)	Ratio $\frac{E}{T}$
4	199	104	0.523
5	257	130	0.506
6	317	156	0.492

the condensation, and there is always present a small depth of water in the bottom of the driers.

43 Owing to the fact that the machines must be operated from the front side, it is the practice to locate the inlet and condensation outlet in the back side of each drier. This arrangement does not permit of a uniform or rapid circulation of steam within the driers. It is the authors' opinion that some system whereby a more rapid circulation of steam were made possible would increase to a considerable extent the heat transfer. This could be accomplished, if it were possible to connect a number of driers in series in such a manner that the circulation of steam would follow the desired temperature gradient.

44 Air and other incondensable vapors, allowed to accumulate in the driers, retard heat transmission and also result in unequal temperature across the drier face. A suitable air valve should be installed to permit of rapid extraction, without allowing steam to pass.

45 Absolutely clean internal surfaces are required to obtain

maximum heat transfer. With engine-driven machines the internal surface of the driers becomes coated with a film of oil, as oil separators do not completely remove all the lubricant from the exhaust. The turbine, however, requires no internal lubrication, and is therefore superior to a steam engine as a prime mover. The external surfaces must also be kept clean of all accumulations of dirt, grease and lint from the paper.

46 Superheated steam, while undoubtedly of great value for prime movers, is not desirable for drying purposes, in that it is similar in its properties to gases and parts with its heat much less readily than saturated steam. No doubt some will contend that superheated steam would tend to become saturated by contact with

TABLE 4 TEST OF DRIER

Working width, inches (deckle)	156
Speed, ft. per min	650
Effective heating surface, sq. ft	4,401
Total surface, sq. ft	8,272
Ratio of effective to total surface	0.532
Weight of paper (lb. per 500 sheets 24 × 36 in.)	32
Lb. paper dried per minute	90
Percentage of moisture in sheet entering driers	71.5
Percentage of moisture in sheet leaving driers	8.0
Lb. moisture evaporated per lb. of dried paper	2.228
Lb. moisture evaporated per minute	200.5
Steam pressure in driers, lb. per sq. in. abs	23.7
Temperature of steam in driers, deg. fahr	248.1
Total heat of steam supplied to driers, B.t.u. per lb	1,164.7
Heat of condensate, B.t.u. per lb	205.1
B.t.u. absorbed by driers per lb. of steam	959.6
Lb. steam per hour to driers	16,822
Lb. steam per minute to driers	280.37
B.t.u. absorbed by driers per minute	269,043
B.t.u. absorbed by driers per lb. of moisture evaporated	1,342

the condensate, but it is thought that this would take place to a very limited extent. The amount of water in contact with the steam varying from time to time would result in variations in temperature within the drier. Some expert papermakers contend that superheated steam produces an inferior quality of newsprint as compared with that dried by saturated steam.

47 The test of the drier part of the machine is given in Table 4. As shown by this table, 269,043 B.t.u. were absorbed by the driers, equivalent to 1342 B.t.u. per pound of moisture evaporated. It will be shown later that the air in passing through the drier sections absorbed 272,837 B.t.u. per min., equivalent to 1361 B.t.u.

per lb. of moisture evaporated. The higher value has been used in subsequent calculations.

48 At the time that the test was conducted to determine the heat absorbed by the driers, data were also obtained giving the temperature gradient throughout the driers and the amount of moisture evaporated per pound of dried paper in each increment of travel, as shown by Fig. 4. By integration the theoretical amount of heat required to evaporate the moisture was determined, as shown by Table 5.

49 It will be noted in this table that 2.228 lb. of water were

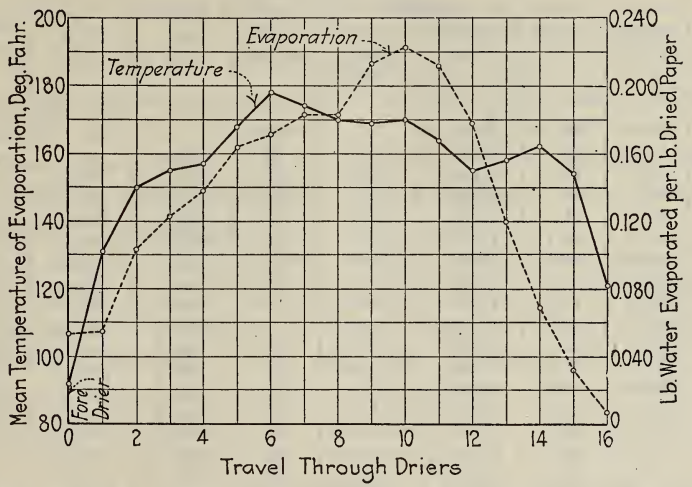

Fig. 4 Curves Showing Temperature of Evaporation and Water Evaporated in Driers

evaporated from the sheet in its complete travel, and that the theoretical heat required to do this work amounted to 2412 B.t.u. Therefore, the theoretical heat required to evaporate one pound of moisture from the sheet, under the conditions as shown, is 1083 B.t.u.

50 In making drier calculations it is well to remember that the only useful work performed is the evaporation of moisture from the sheet, and that therefore the theoretical amount, as determined by the conditions of the test, divided by the heat supplied, must give the efficiency of the driers.

51 Based on the figure of 1342 B.t.u., determined by the

drier test, the efficiency is 80.7 per cent, while on the basis of 1361 B.t.u., derived from the air test, it is 79.6 per cent.

52 To dry paper efficiently the temperature conditions throughout the drier sections must be closely regulated. In the past, temperature control has been obtained by means of a thermostatic valve, which was dependent upon the temperature existing in one drier only. While this has given good results, the latest

TABLE 5 THEORETICAL HEAT REQUIREMENTS OF DRIERS PER LB. OF PAPER

Drier No.	Moisture leaving drier	% stock in sheet	Lb. water per 0.92 lb. stock	Lb. water evaporated in travel through drier	Mean temp. of evap., deg. fahr.	Latent heat	B.t.u.	Heat of Liquid
	71.5 [1]	28.5 [1]	2.308 [1]	.....	80 [1]	.....	.....	.....
Fore	71.0	29.0	2.254	0.054	92	1040.0	56.16	0.65
1	70.5	29.5	2.199	0.055	131	1018.2	56.00	2.81
2	69.5	30.5	2.096	0.103	150	1007.4	103.76	7.21
3	68.2	31.8	1.973	0.123	155	1004.5	123.55	9.23
4	66.6	33.4	1.835	0.138	157	1003.4	138.47	10.63
5	64.5	35.5	1.671	0.164	168	997.0	163.51	14.43
6	62.0	38.0	1.500	0.171	178	991.1	169.48	16.76
7	59.0	41.0	1.317	0.183	174	993.5	181.81	17.20
8	55.2	44.8	1.1336	0.1834	170	995.8	182.63	16.47
9	50.0	50.0	0.9200	0.2136	169	996.4	212.83	19.05
10	43.1	56.9	0.6969	0.2231	170	995.8	222.16	20.07
11	34.5	65.5	0.4846	0.2123	164	999.3	212.15	17.81
12	25.0	75.0	0.3067	0.1779	155	1004.5	178.70	13.34
13	16.9	83.1	0.1871	0.1196	158	1002.8	119.93	9.33
14	11.3	88.7	0.1183	0.0688	162	1001.1	68.88	5.64
15	8.6	91.4	0.0866	0.0317	154	1005.1	31.86	2.35
16	8.0	92.0	0.0800	0.0066	121	1023.9	6.76	0.27
Total	....	....	.....	2.2280	....	.....	2228.64	183.25

[1] Condition of sheet entering the drier.

practice calls for sectional control, which has improved the ease of regulation.

53 Without suitable experimental equipment it is very difficult to proportion the total heat transfer into that quantity absorbed by the sheet and that acquired by the surrounding air. It is thought that these quantities, once determined, would be of great assistance to the designers of paper machines, as at present, so far as the authors are aware, all calculations are based on an arbitrary figure.

HEATING AND VENITLATING

54 Ventilating air supplied to a machine room may be considered as serving three distinct purposes:

1 It maintains such temperatures and humidities within the room that the operators may perform their work in comfort
2 During the period of low outside temperature it supplies sufficient heat to replace the radiation loss from the building
3 It acts as a carrier for the moisture liberated by the machine, so that this moisture is conveyed to the outside air without condensation taking place in the room.

55 The quantity of air supplied to a machine room depends to a large degree upon the desired machine-room temperature and humidity; and it will be found for the ordinary room, all other conditions considered, that the quantity of air insuring greatest economy is largely exceeded. This practice is only adhered to because of the resulting comfort of the operators.

56 Even under the best conditions the temperatures found in a machine room are quite high. Particularly is this true in the vicinity of the roof and in monitors, where the average temperature may vary from 80 to 100 deg. fahr. Further, the walls of a modern machine room contain a large glass area, resulting in a considerable heat transfer to the outside air. The radiation loss for the building under discussion may be calculated from the formula $y = 5.1 - 0.0583x$, where y is the radiation loss per pound of air in B.t.u., and x the outside or initial air temperature in deg. fahr. This formula is based on an exhaust-fan discharge of 5422 lb. of air per minute, and it shows that the radiation loss from the building makes a very appreciable demand upon the heating load for ventilation.

57 The air supplied for carrying away the moisture from the machine may be considered as ventilating the machine, apart from the ventilation of the room. This air, in its passage through the machine room, acquires from the wet end a certain amount of moisture, but no heat, and from the driers, both moisture and heat. It must, therefore, when expelled from the room, contain this moisture and heat, together with that which it contained when coming in contact with the machine. Provided, therefore, that the initial and final conditions of the air are known, and also its

quantity per unit of time, the amount of moisture and heat liberated by the machine can be readily calculated.

58 In order to determine these quantities, so that they could be used as constants in further calculations, a test, previously referred to, was conducted on a still day in summer when the machine-room windows were all open and air of a uniform temperature was supplied to the room. The advantage of making this test in summer rather than in winter is apparent when it is considered that in winter considerable leakage takes place through openings from adjoining buildings and from the outside air, which renders it difficult to arrive at a representative value of the temperature and relative humidity of the incoming air. Also the outside temperature at the time being 83.3 deg. fahr., the radiation loss from the building could be neglected without introducing any appreciable error. The results of this test are shown in Table 6.

59 In making this test the discharge of hot air from each of the stacks was determined by dividing their rectangular cross-section into 24 equal divisions, and the velocity at the center of each of these divisions was determined by means of an anemometer. Wet- and dry-bulb temperatures were taken at the same time, and the discharge as computed in cubic feet per minute was converted into pounds per minute. Carrier's psychometric charts were used to determine the moisture per pound of dry air and the total heat above 0 deg. fahr. per lb. of air. These charts were of great assistance in simplfying the calculations.

60 It is thus seen that the quantity of heat given up by the driers per minute as calculated on the ventilating-air basis, checked that as calculated on a steam basis within 1.5 per cent.

61 The difference between the moisture acquired by the air, 204 lb., and that liberated by the driers, 200.5 lb., can, in the opinion of the authors, be reasonably taken as the amount of moisture acquired by the ventilating air from the wet end of the machine, and in this case amounted to 3.5 lb., or less than 2 per cent of the total.

62 When, therefore, the machine is operating at 650 ft. per min. and 5422 lb. of air are expelled by the exhaust fans, the heat given up by the machine per minute is 272,837 B.t.u. and the moisture liberated in the same time is 204 lb.

63 If then values are assumed for the initial air condition,

it is possible, by utilizing the above figures, to determine the final moisture and heat content of the air per pound, and by comparing the heat content of air containing this moisture with that of air containing the same moisture and at saturation, or 80 per cent relative humidity, it is possible to determine to what degree the initial outside air must be heated before entering the machine room, in order that it may function properly.

64 It must be borne in mind that the above statement presupposes that for low outside-air temperatures the ventilating air is preheated before admission to the machine room. Consequently,

TABLE 6 VENTILATING-AIR TEST

	Stack No.	Dry-bulb temp., deg. fahr.	Wet-bulb temp., deg. fahr.	B.t.u per lb. of air above 0° F.	Grains moisture per lb. dry air.	Lb. air per min.	Lb. dry air per min.	Lb. moisture in air.	B.t.u above 0° F. contained in air.
Final air in stack.....	1	110	108.5	85	396	1788	1692	96	151,980
	2	118	107.0	82	360	1766	1680	86	144,812
	3	121	113.0	95	445	1868	1756	112	177,460
Total.....	....	....	....	....	....	5422	5128	294	474,252
Initial air entering room....	....	83.3	76	38.6	123.3	5218	5128	90	201,415
Difference in moisture and heat content between final and initial air.....								204	272,837

upon reaching the machine its temperature will be practically the same as applied when the test was made, and therefore the heat transfer from the driers, and the moisture absorbed from the wet end, will not, for practical purposes, be changed.

65 Table 7 has been prepared on a basis of initial air at 70 per cent relative humidity. From this table Fig. 5 has been plotted, which serves the purpose of estimating the machine heating load for various outside temperatures.

66 By referring to Fig. 5, it is seen that for the ventilation of the machine only, there is some critical temperature, in this case 66 deg. fahr., below which the outside or initial air must be preheated before coming in contact with the machine. In practice, however, this critical temperature would be slightly higher,

as it is not common practice to expel the air from the machine room at a relative humidity (R. H.) as high as 100 per cent. If the final air had a relative humidity of 80 per cent, the critical temperature would be 78 deg. fahr.

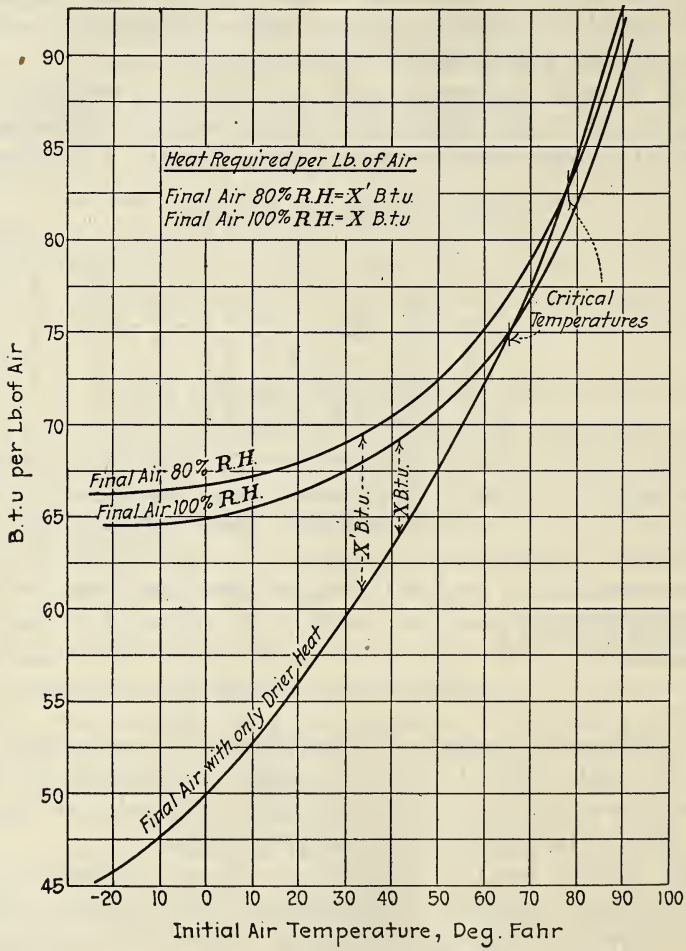

Fig. 5 Curves Showing Heat Required for Paper-Machine Ventilation

67 By combining the quantity of heat per pound of air required by the machine with that necessary to replace the radiation loss from the building, the total heat per pound of air required for the purpose of ventilation can be determined.

TABLE 7 SHOWING CHANGE IN AIR CONDITION WHILE PASSING THROUGH THE MACHINE

Initial Air Condition								Absorbed From Machine			Final Air Condition							
Temperature °F.	% Relative Humidity	Gr. Moisture per lb. of dry air	Heat above 0° F. per lb. of air, B.t.u.	Lb. of air per min.	Lb. dry air per min.	Total moisture, lb.	Total heat, B.t.u.	Moisture from wet end, lb.	Moisture from driers, lb.	Heat from driers, B.t.u.	Total air, lb.	Total Moisture, lb.	Dry air, lb.	Gr. moisture per lb. of dry air	Heat above 0° F. in total air, B.t.u.	B.t.u. per lb. of air	B.t.u. per lb of air, 100% R.H. and same moisture	B.t.u. per lb. of air, 80% R. H. and same moisture
−20	0	0	−4.8	5218	5218	0	−25046	3.5	200.5	272,837	5422	204	5218	274	247,791	45.7	64.5	66.3
+10	70	6.5	2.5	5218	5213	5	13045	3.5	200.5	272,837	5422	209	5213	280	285,882	52.7	65.5	67.1
30	70	17.0	9.75	5218	5205	13	50876	3.5	200.5	272,837	5422	217	5205	292	323,713	59.6	67.5	69.0
40	70	25.6	13.6	5218	5199	19	70965	3.5	200.5	272,837	5422	223	5199	301	343,802	63.3	69.0	70.5
50	70	37.3	17.9	5218	5190	28	93402	3.5	200.5	272,837	5422	232	5190	313	366,239	67.7	70.7	72.4
60	70	54.0	22.8	5218	5178	40	118970	3.5	200.5	272,837	5422	244	5178	330	391,807	72.2	73.3	75.2
70	70	76.0	28.3	5218	5162	56	147669	3.5	200.5	272,837	5422	260	5162	353	420,526	77.5	76.8	79.0
80	70	107.0	35.6	5218	5140	78	185761	3.5	200.5	272,837	5422	282	5140	384	458,598	84.5	82.3	84.2
90	70	149.2	44.2	5218	5110	108	230636	3.5	200.5	272,837	5422	312	5110	427	503,473	92.7	89.4	91.6

68 The important factor which influences the ventilating requirements of a paper mill is the amount of moisture liberated by the machine per unit of time. This in turn is dependent upon:

1 Speed of the machine
2 Moisture in sheet entering the driers
3 Moisture in sheet leaving the driers
4 Moisture absorbed from the wet end.

69 The old practice, therefore, of specifying paper-mill ventilation on a basis of so many minutes air change, without due regard to the moisture liberated, is not correct, as it is obvious that the amount of moisture liberated in two machine rooms of the same size might be entirely different.

70 Much of late has been written regarding the proper distribution of air in the machine room, but it is not the intention of the authors to discuss this further, except to state that, in their opinion, the greater part of the ventilating air should be delivered in the vicinity of the driers and the wet end, so that the moisture may be confined at its source. Further, that sufficient warm dry air should be supplied to the roof and all pockets, such as monitors, in order that the moisture-laden air shall not become chilled below the dewpoint, causing drip.

71 The amount of heat loss to the process by the expelled ventilating air is in all probability the greatest unpardonable waste in the manufacture of paper.

72 By referring to Table 7, it can be seen that, with initial air at 10 deg. fahr. and of 70 per cent relative humidity, the total heat above zero deg. fahr. is 2.5 B.t.u. per lb., and the heat content of a pound of final air at 100 per cent relative humidity is 65.5 B.t.u. above 0 deg. fahr., the difference in the heat content per pound of air being 63 B.t.u.

73 Only 12.8 B.t.u. per lb., however, have to be furnished in order to preheat the incoming air, the balance of this difference being supplied by the driers. It is here that an economizer could be used to great advantage in reclaiming from the expelled air enough heat to condition the incoming air.

74 Owing to the high relative humidity of the outgoing air, a few degrees' drop in temperature in the economizer would be sufficient to supply heat necessary to preheat the incoming air,

with the result that no live or exhaust steam would be required for ventilation.

75 The saving in operating cost which would result in winter from such an installation is apparent when it is considered that, for the conditions under consideration, at least 5900 lb. of dry steam, at 10 lb. per sq. in. pressure, would be required per hour for

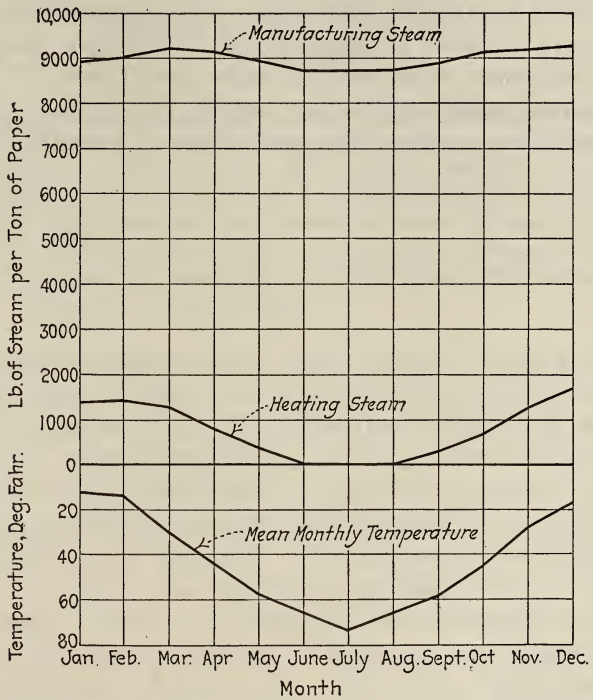

Fig. 6 Curves Showing Temperature and Steam for Manufacturing and Heating, by Months

preheating the ventilating air when the outside temperature was 10 deg. fahr. and the relative humidity 70 per cent.

76 Fig. 6 shows for the period of one year the manufacturing and heating load in pounds of steam per ton of product, also the mean monthly temperature which prevailed. It is interesting to note the relative importance of the heating load, and it can be readily realized that the manufacturing cost could be considerably reduced if, by the installation of an economizer, this were done away with.

CONCLUSION

77 This paper does not make a complete survey of the subject under discussion, but the authors, in seeking to collect material for the subject-matter, have been impressed with the lack of detailed information which is available. It may be that some mills have studied their problems in great detail, but if so, the information is treated as strictly confidential. It can be understood that, once a paper machine is installed, the owners are reluctant to decrease production while making experimental tests. Therefore, paper-machine manufacturers are unable, in certain respects, to procure reliable operating data upon which to base their designs.

No. 1864

TORSION OF CRANKSHAFTS

By S. Timoshenko,[1] Philadelphia, Pa.
Non-Member

In order to apply the dynamics of elastic systems to the designs of engines it is necessary to determine the torsional properties of the crankshaft. In the present paper the author considers the case of a crankshaft with a single throw and establishes the mathematical relations for such a case. He investigates three conditions of such a crankshaft: (a) *no constraint, corresponding to ample clearance in the bearings;* (b) *complete constraint, corresponding to no clearance in the bearings; and* (c) *partial constraint, corresponding to ample clearance in the halves of the bearings nearest the web and no clearance in the other halves.*

THE time has passed when engine design comprises merely the application of statics, and more attention is now paid to the fact that in various types of engines there is not even an approximation of a state of equilibrium. If there are variations in the torque and in the magnitudes of the force, they are necessarily cyclic, and as such they produce cyclic changes in the state of motion and deformation of the parts. The problem is no longer one of statics, but concerns the dynamics of an elastic system. Variations in the state of motion are associated with inertia forces, and the stresses in the various parts of the engine are no longer due to the actually impressed forces alone. Obviously, calculations which consider only the impressed forces must lead to erroneous results, and frequently a break occurs where the designer thought there was an ample margin of strength. Moreover cyclic changes in the state of deformation of a crankshaft, for example, bring about a cyclic change in the orientation of the various cranks, and the balance of the reciprocating parts, which may exist without such changes, is totally destroyed. With reciprocating engines the cyclic changes in the state of motion, referred to above, concern principally changes of the velocity of rotation. In order to apply the dynamics of

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elastic systems to the type of engine just mentioned, it is necessary to determine the torsional properties of the crankshaft. This is the object of the present paper. This paper deals with the simpler cases and will be followed by other papers on the same subject. In addition to the derivation of the various formulas, it is shown how the amount of constraint affects the torsional properties.

2 On account of the very complex structure of reciprocating engines, the calculation of the torsional vibrations in their crankshafts is impossible without making some simplifying assumptions; i.e., that certain parts, such as shafts, are considered to be elastic, and others, such as flywheels, armatures of generators, etc., are considered to be absolutely rigid. With these assumptions, the engine may be reduced to a system of flywheels situated on a shaft of uniform diameter. This shaft is called the "equivalent shaft"; its diameter may be chosen arbitrarily, but its length between each two flywheels must be such that it is equivalent as to torsion to the actual shaft between corresponding parts of the actual engine. That is to say, a given twisting moment M must produce the same torsional deformation in both.

3 The length of the equivalent shaft is called the reduced length l_0. If we denote by the torsional rigidity C of a shaft, the product of the modulus of shear G into the quantity which measures its resistance to twist, then by elementary mechanics we have for the twist δ of a straight shaft of length l of whatsoever cross-section (provided it is uniform),

$$\delta = \frac{Ml}{C} \quad \text{[1]}$$

4 In the above, as well as in the following, angles will be expressed in radians and all other units in inches and pounds.

5 For a circular cross-section,

$$C = G\theta \quad \text{[1a]}$$

wherein $\theta = \frac{\pi}{32} d^4$ is the polar moment of inertia of the cross-section.

6 Similarly, we have for the equivalent shaft:

$$\delta = \frac{Ml_0}{C_0} \text{ and } C_0 = G\theta_0 \quad \text{[2]}$$

from which

$$l_0 = \frac{C_0 l}{C} = \frac{\theta_0 l}{\theta} \quad \text{[3]}$$

7 As the foregoing shows, the calculation of the reduced lengths of the equivalent shaft has no difficulty for those portions of shaft in the actual engine which are straight. When the actual shaft is a crankshaft, however, the situation becomes very much more involved, since it becomes necessary to determine the angular deformation δ brought about by a twisting moment M. Owing to the complex geometric shape of a crankshaft, these calculations can be accomplished approximately only. Various assumptions will have to be made, of which some are only roughly true. It is assumed that the throw is built up of component shapes, whose deformations are totally independent of each other. Further, it is necessary to make certain assumptions about the nature and amount of constraint in the bearings. In the present paper only the single-throw crankshaft (Fig. 1) will be considered, and therefore the very considerable influence on it of the neighboring cranks will be neglected. Calculations will be made with three degrees of constraint.

DEFINITIONS AND SYMBOLS

8 Fig. 1 shows the conventional type of a single-cylinder two-bearing crankshaft. In addition to the dimensions shown thereon, the following is a list of the definitions and principal symbols used in the paper:

9 Torsional rigidity, generally denoted by C, is the product of the modulus of shear G into the quantity corresponding to its resistance to twist. For a circular cross-section, the latter is the polar moment of inertia of the cross-section in respect to the axis of twist. For a rectangular cross-section, the quantity referred to is more complex. The formula usually used is only approximately true.

10 Flexural rigidity is similarly the product of Young's modulus into the quantity I corresponding to the resistance of the cross-section against bending. The latter is always the equatorial moment of inertia of the cross-section in respect to the axis of bending.

$C_0 = G\theta_0 = \frac{\pi}{32} d_0^4 G$ is the torsional rigidity of the equivalent shaft, d_0 being its diameter, l_0 its length.

$C_1 = G\theta_1 = \frac{\pi}{32} d_1^4 G$ is the torsional rigidity of the journal.

$C_2 = G\theta_2 = \frac{\pi}{32} d_2^4 G$ is the torsional rigidity of the crankpin.

$C_3 = G\theta_3$ is the torsional rigidity of the web in respect to twist around O-O. Owing to the junction of journal and pin to the web, the latter's cross-section is not clearly defined, but in order to allow for local stresses at these junctions, at least in some measure, the assumption is made that the cross-section is a rectangle with sides r and c, whence —

$$\theta_3 = \frac{c^3r^3}{3.6(c^2 + r^2)}$$

$C_3' = G\theta_3'$ is the torsional rigidity of the web with respect to

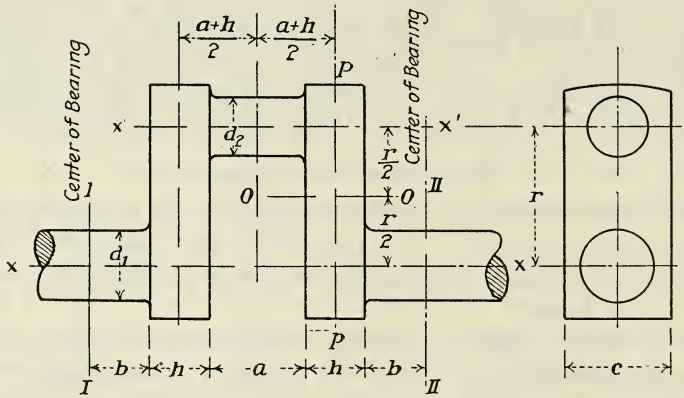


FIG. 1 CONVENTIONAL TYPE OF SINGLE-CYLINDER, TWO-BEARING CRANKSHAFT

twist around p-p. The cross-section is rectangular with sides h and c, whence —

$$\theta_3' = \frac{c^3h^3}{3.6(c^2 + h^2)}$$

$B_1 = EI_1 = \frac{\pi}{64} d_1^4 E$ is the flexural rigidity of the journal.

$B_2 = EI_2 = \frac{\pi}{64} d_2^4 E$ is the flexural rigidity of the crankpin.

$B_3 = EI_3 = \frac{hc^3}{12} E$ is the flexural rigidity of the web against bending in the plane through p-p perpendicular to the plane of the drawing of Fig. 1.

$F_1 = \frac{\pi d_1^2}{4}$ is the area of the cross-section of the journal

$F_2 = \frac{\pi d_2^2}{4}$ is the area of the cross-section of the pin

$F_3' = h \times c$ is the area of the cross-section of the web taken on O-O

$F_3 = r \times c$ is taken to be the cross-section of the web on line p-p. This quantity is used in the calculation for the deformation due to shear in the plane through p-p perpendicular to the plane of Fig. 1. Similarly, as in the case of torsion, around O-O the cross-section is not clearly defined and $r \times c$ is used to allow for stresses at the junction of pin and journal to the web.

CASE I NO CONSTRAINT, CORRESPONDING TO AMPLE CLEARANCE IN THE BEARINGS

11 Fig. 2a is a diagrammatic representation of the throw before deformation with a twisting moment M applied at the middle cross-section of the journals. Fig. 2b is a side view of the deformed crankshaft. The total twist consists of the sum of the deformations of the portions b (see Fig. 1) of the journals, of the crankpin and of the two webs.

12 All these parts are here subject to torsion only around the axes x-x; x'-x' and O-O, except the webs, which also suffer bending. Now if δ_1, δ_2, δ_3 are the angles of twist of the above parts, we have

$$\delta_1 = \frac{Mb}{C_1};\ \delta_2 = \frac{Ma}{C_2};\ \delta_3 = \frac{Mh}{C_3} \quad \text{[4]}$$

13 Further, the bending of the web produces the angular displacement δ_4, equal to the angle between the tangents to the curve of flexure. Since the bending moment is equal to the twisting moment M and is constant, the curve of flexure of the web is a circle. Considering it a beam of length r, we have —

$$\delta_4 = \frac{Mr}{B_3} \quad \text{[4a]}$$

14 The total relative angular displacement δ, measured at the middle section of the journals, will be —

$$\delta = 2\delta_1 + \delta_2 + 2\delta_3 + 2\delta_4$$

and substituting for δ_1, δ_2, δ_3 and δ_4, we obtain —

$$\delta = M\left\{\frac{2b}{C_1}+\frac{a}{C_2}+\frac{2h}{C_3}+\frac{2r}{B_3}\right\}$$

15 For the equivalent shaft we have

$$\delta = \frac{Ml_0}{C_0}$$

whence

$$l_0 = \frac{2bC_0}{C_1}+\frac{aC_0}{C_2}+\frac{2hC_0}{C_3}+\frac{2rC_0}{B_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

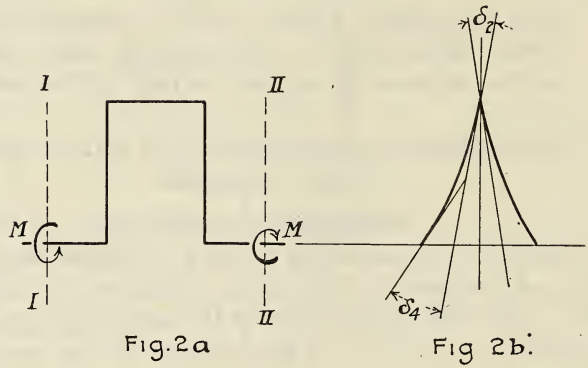


FIG. 2*a* DIAGRAMMATIC REPRESENTATION OF SINGLE-THROW CRANKSHAFT BEFORE DEFORMATION

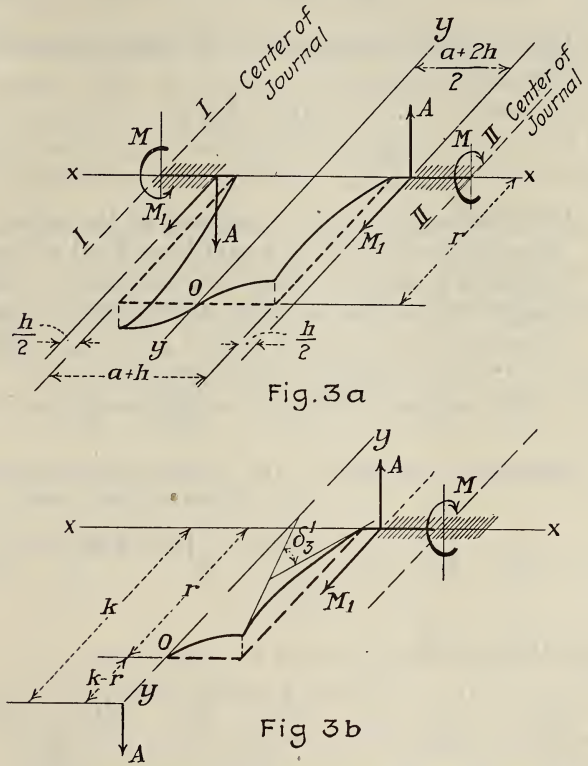
FIG. 2*b* DIAGRAMMATIC REPRESENTATION OF SIDE VIEW OF DEFORMED CRANKSHAFT

CASE II CONSTRAINT AT BEARINGS COMPLETE, CORRESPONDING TO NO CLEARANCE

16 The deformed crank is shown diagrammatically in Figs. 3*a* and 3*b*. The constraint gives rise to a force A and a moment represented by the vector M_1 at each of the journals. The forces A and moments M_1 act in a plane through the axis of the journals perpendicular to the plane of the throw.

17 On the assumption that the deformations are small, we derive from the reciprocal theorem that there will be no forces in the plane of the throw, for forces in the plane of the throw cannot produce torsional displacements corresponding to the twisting moment M, and reciprocally M cannot cause displacements, nor can it give rise to forces, in the plane just mentioned.

18 By symmetry we conclude that the middle point O of the crankpin becomes a point of inflection, and if so, there are in this cross-section shearing stresses only. Thus, if we remove half of the throw, as in Fig. 3*b*, the effect of the removed half on the remainder is a single force. And since the remaining portion is in equilibrium,



FIGS. 3*a* AND 3*b* DIAGRAMMATIC REPRESENTATION OF DEFORMED CRANK — CASE II

this force must be equal and opposite to A and its point of application will be at a certain distance k from the axis x-x.

19 Because of the equilibrium there also must be no moments. Taking them round x-x and y-y, this condition requires

$$M_1 = \frac{A(a+2h)}{2} \text{ and } Ak = M \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

20 The angular displacement of the throw measured at the sections I-I and II-II will consist of the following components:

1 The torsional displacement of the journals —

$$2\delta_1 = \frac{2Mb}{C_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

2 The torsional displacement of the crankpin —

$$\delta_2 = \frac{A(k-r)a}{C_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7a]$$

3 The torsional deformation of the webs due to twist around *O-O* (Fig. 1) —

$$2\delta_3 = \frac{2A\left(k-\frac{r}{2}\right)h}{C_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7b]$$

4 The displacement due to bending of the webs. In the present case the web is a cantilever fixed at the journal and with a force A acting at a distance k. Taking the length of the web to be r, the angle δ_3' between the tangents to the curve of flexure at its ends will be —

$$2\delta_3' = \frac{Ar^2}{B_3} + \frac{2A(k-r)r}{B_3} = \frac{2Ar}{B_3}\left(k-\frac{r}{2}\right) \quad . \quad . \quad . \quad . \quad [7c]$$

21 Summing up, we get for the angular displacement of the throw —

$$\delta = \frac{2Mb}{C_1} + \frac{A(k-r)a}{C_2} + \frac{2A\left(k-\frac{r}{2}\right)h}{C_3} + \frac{2Ar\left(k-\frac{r}{2}\right)}{B_3}$$

or since by [6] $A = \frac{M}{k}$,

$$\delta = \frac{2Mb}{C_1} + \frac{M(k-r)a}{kC_2} + \frac{2M\left(k-\frac{r}{2}\right)h}{kC_3} + \frac{2Mr\left(k-\frac{r}{2}\right)}{kB_3} \quad [7d]$$

For the equivalent shaft we have —

$$\delta = \frac{Ml_0}{C_0}$$

whence by equating with [7d] we obtain —

$$l_0 = \left\{\frac{2bC_0}{C_1} + a\left(1-\frac{r}{k}\right)\frac{C_0}{C_2} + 2h\left(1-\frac{r}{2k}\right)\frac{C_0}{C_3} + 2r\left(1-\frac{r}{2k}\right)\frac{C_0}{B_3}\right\} \quad . \quad [8]$$

22 From the above we see that the reduced length of the throw is completely determined if we know the fraction $\frac{r}{k}$. For

$A = 0$ or $k = \infty$ we must have the case of no constraint and, indeed, by placing $k = \infty$ in [8] we obtain [5]. The effect of the constraint is therefore measured by the quantity k. And in the special case where $2b = a$ and $C_1 = C_2$ we can immediately compare the case of a complete constraint with no restraint and find that the constraint decreases the reduced length in the ratio of 1 to $\left(1 - \frac{r}{2k}\right)$.

23 The calculation of k proceeds as follows: Imagine the half of the throw (Fig. 3*b*) before deformation rotated around x-x through one-half the angle δ of [7d], and the cross-section at *II-II* thereupon clamped. Under this supposition, the point O will be displaced perpendicular to the initial plane of the throw the distance $\frac{r\delta}{2}$. If we then apply the force A and the moment M, the displacement just mentioned must vanish, since O, according to the foregoing, has no displacement perpendicular to the initial plane of the crankshaft.

24 We therefore proceed to calculate the displacements of the point O brought about by the various forces and equate their sum to $\frac{r\delta}{2}$.

25 These displacements are:

1 The deflection of O, or —

$$\frac{Aa^3}{24B_2} \quad \ldots\ldots\ldots\ldots \quad [9]$$

due to the flexure of the crankpin in a plane perpendicular to the initial plane of the throw. It will be noted that we have here the case of a cantilever of length $\frac{a}{2}$, acted upon by a force A at its end

2 The deflection of O, or —

$$\frac{\gamma Aa}{2F_2G} \quad \ldots\ldots\ldots\ldots \quad [9a]$$

due to the shearing force A in the crankpin, wherein γ is a coefficient taken to be 1.2

3 The displacement of the point O —

$$\frac{A(a+h)^2r}{4C_3'} \quad \ldots\ldots\ldots\ldots \quad [9b]$$

due to twisting of the web around p-p, brought about

by the force A acting on a lever $\frac{(a+h)}{2}$. In the above the length of the web, subject to torsion, has been taken equal to r. This is approximately true in such cases where there is a considerable portion of the web free between the junctions of journal and pin, that is to say, where $\frac{(d_1+d_2)}{2}$ is small as compared with r. For heavy short-stroke engines, the assumption is far from true.

4 The displacement of O —

$$\frac{Ar^2(3k-r)}{6B_3} \quad \ldots\ldots\ldots\ldots \quad [9c]$$

due to bending of the web in the plane through p-p and perpendicular to the plane of Fig. 1.

5 The displacement of O

$$\frac{\gamma Ar}{F_3'G} \text{ and } \frac{\gamma Ah}{F_3G} \quad \ldots\ldots\ldots\ldots \quad [9d]$$

due to shear in the web, brought about by the shearing force A.

6 The displacement of O —

$$\frac{r\delta_3}{2} = \frac{A\left(k-\frac{r}{2}\right)hr}{2C_3} \quad \ldots\ldots\ldots\ldots \quad [9e]$$

due to twist of the web around O-O.

7 The displacement of O —

$$r\delta_1 = \frac{Mbr}{C_1} \quad \ldots\ldots\ldots\ldots \quad [9f]$$

due to the twist of the portion b of the journal.

26 Equating the sum of the above displacements to $\frac{r\delta}{2}$, we find —

$$\frac{r\delta}{2} = \frac{Aa^3}{24B_2} + \frac{\gamma Aa}{2F_2G} + \frac{A(a+h)^2r}{4C_3'} + \frac{Ar^2(3k-r)}{6B_3} + \frac{\gamma Ar}{F_3'G} + \frac{\gamma Ah}{F_3G}$$
$$+ \frac{A\left(k-\frac{r}{2}\right)hr}{2C_3} + \frac{Mbr}{C_1} \quad \ldots \quad [10]$$

Substituting therein for δ from [7d] and placing $A = \frac{M}{k}$, we obtain —

$$k = \frac{\dfrac{r(a+h)^2}{4C_3'} + \dfrac{ar^2}{2C_2} + \dfrac{a^3}{24B_2} + \dfrac{r^3}{3B_3} + \dfrac{hr^2}{4C_3} + \dfrac{\gamma a}{2F_2G} + \dfrac{\gamma r}{F_3'G} + \dfrac{\gamma h}{F_3G}}{\dfrac{ar}{2C_2} + \dfrac{r^2}{2B_3} + \dfrac{hr}{2C_3}} \quad [10a]$$

27 This equation permits the calculation of k whence l_o follows from [8]. It is at variance with the equation given by Geiger (Ueber Verdrehungsschwingungen von Wellen), for which reason, and as a check on the above results, the derivation of k and l by means of Castigliano's theorem has been repeated (see Appendix). This method is particularly useful in more complicated cases where the deformation of the throw is not as easily seen as in the preceding case.

CASE III PARTIAL CONSTRAINT

28 It is here supposed that there is ample clearance in the halves of the bearings nearest the web, and no clearance in the other half. In this manner we will now have bending, as well as torsion, in the halves of the journal next to the webs, a condition which no doubt prevails in practice.

29 In the present case [7d] remains as before but [10] has to include the bending of the half of the journal. We here obtain —

$$\frac{r\delta}{2} = \frac{Aa^3}{24B_2} + \frac{\gamma Aa}{2F_2G} + \frac{A(a+h)^2r}{4C_3'} + \frac{Ar^2(3k-r)}{6B_3} + \frac{\gamma Ar}{F_3'G} + \frac{\gamma Ah}{F_3G}$$
$$+ \frac{A\left(k-\frac{r}{2}\right)hr}{2C_3} + \frac{Mbr}{C_1} + \frac{Ab^3}{3B_1} + \frac{A(a+2h)b^2}{2B_1} + \frac{Ab(a+2h)^2}{4B_1} + \frac{\gamma Ab}{F_1G} \quad [11]$$

The ninth to eleventh terms in this equation are due to the bending of the journal, considered as a beam of length b, fixed at the journal with a force A at a distance of $\frac{(a+2h)}{2}$ (see Fig. 4c) from the end. The twelfth term of [11] is the deformation at O due to the shear of the journal brought about by the force A.

30 From [6] and [11] we derive —

$$k = \frac{\dfrac{r(a+h)^2}{4C_3'} + \dfrac{ar^2}{2C_2} + \dfrac{a^3}{24B_2} + \dfrac{r^3}{3B_3} + \dfrac{hr^2}{4C_3} + \dfrac{\gamma a}{2F_2G} + \dfrac{\gamma r}{F_3'G} + \dfrac{\gamma h}{F_3G}}{\dfrac{ar}{2C_2} + \dfrac{r^2}{2B_3} + \dfrac{hr}{2C_3}}$$
$$+ \frac{b^3}{3B_1} + \frac{b^2(a+2h)}{2B_1} + \frac{b(a+2h)^2}{4B_1} + \frac{\gamma b}{F_1G} \quad . \; . \; . \quad [12]$$

NUMERICAL EXAMPLE

31 Let $d_1 = d_2 = 10.25$ in.; $a = 13$ in.; $r = 11$ in.; $b = a/2 = 6.5$ in.; $h = 5.5$ in.; $c = 14$ in.

whence —

$$B_1 = B_2 = \frac{\pi \times 10.25^4 E}{64} = 542E; B_3 = \frac{5.5 \times 14^3 E}{12} = 1258E$$

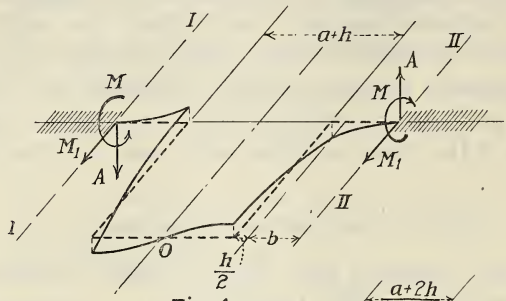


Fig. 4a

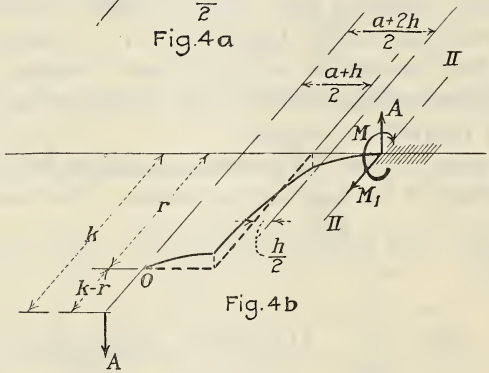


Fig. 4b

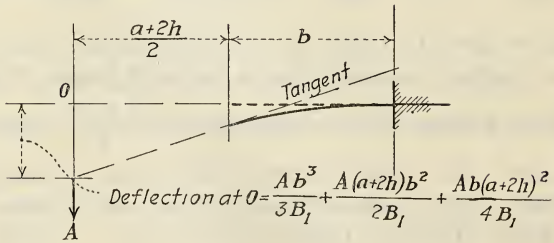


Fig 4c

FIGS. 4*a*, 4*b*, 4*c* DIAGRAMMATIC REPRESENTATION OF CRANKSHAFT — CASE III

$$C_1 = C_2 = \frac{\pi \times 10.25^4}{32} G = 1085G; \; C_3' = \frac{14^3 \times 5.5^3}{3.6(14^2 + 5.5^2)} G = 562G$$

$$C_3 = \frac{14^3 \times 11^3}{3.6(14^2 + 11^2)} G = 3200G$$

We take $\frac{E}{G} = 2.6$.

The diameter of the equivalent shaft we take equal to the diameter of journal, whence in this example $C_0 = C_1 = C_2$.

32 With the above we have, when there is no constraint (Case I) from [**5**]

$$l_0 = 37 \text{ in.}$$

For the case of a complete constraint (Case II), we obtain from [**10a**] —

$$k = 31.6 \text{ in.}$$

and since in the present case $2b = a$, the complete constraint reduces l_0 in the ratio —

$$1 : \left(1 - \frac{r}{2k}\right) = 1 : (1 - 0.174)$$

that is, the reduced length is reduced by 17.4 per cent.

33 In the case of a partial constraint (Case III), we obtain from [**12**] —

$$k = 44.2 \text{ in.}$$

and the reduced length, as compared with the case of no constraint, is reduced in the ratio —

of $1 : \left(1 - \frac{11}{2 \times 44.2}\right)$, that is, by 12.5 per cent.

34 These examples plainly show the effect of the constraint at the bearings. The more complete it is, the stiffer the shaft and the shorter the reduced length. The calculations further show that an increase in the diameters of journal and pin and an increase in the length of the pin cause an increase of k and thereby a reduction of the effect of the constraint and reduced bearing pressures A. On the other hand, an increase in the thickness of the web brings about a smaller length k and a corresponding augmentation of the bearing reactions.

APPENDIX

35 In applying the theorem of Castigliano to Case II (throw with complete restraint), the potential energy of each of the component parts is determined as follows:

1 The potential energy of bending of the crankpin —

$$V_1 = 2\int_0^{\frac{a}{2}} \frac{A^2x^2dx}{B_2} = \frac{A^2a^3}{24B_2}$$

2 The potential energy of torsion of the crankpin —

$$V_2 = \frac{(M - Ar)^2a}{2C_2}$$

3 The potential energy due to shear in the crankpin —

$$V_3 = \frac{\gamma A^2a}{2F_2G}$$

4 The potential energy of bending of the web —

$$V_4 = 2\int_0^r \frac{(M - ay)^2}{2B_3}\,dy = \frac{M^2r - MAr^2 + A^2\frac{r^3}{3}}{B_3}$$

5 The potential energy of torsion of the webs brought about by the force A with the lever $\frac{(a+h)}{2}$ —

$$V_5 = \frac{A^2(a+h)^2r}{4C_3'}$$

6 The potential energy of torsion of the webs about the axis O-O (Fig. 2), the twisting moment being $M - \frac{Ar}{2}$ —

$$V_6 = \frac{\left(M - \frac{Ar}{2}\right)^2h}{C_3}$$

7 The potential energy due to the shearing force A in the webs —

$$V_7 = \frac{\gamma A^2r}{F_3^1G} + \frac{\gamma A^2h}{F_3G}$$

8 The potential energy due to torsion of the journals —

$$V_8 = \frac{M^2b}{C_1}$$

36 The potential energy of the entire throw is —

$$V = V_1 + V_2 + \ldots V_8$$

In the above we consider M as given and A as the variable to be determined. On the supposition, then, of a complete constraint at the bearings, the force A at its point of application brings about no displacement, whence by Castigliano's theorem —

$$\frac{\partial V}{\partial A} = 0 = \frac{\partial(V_1 + V_2 + \ . \ . \ . \ V_8)}{\partial A}$$

Substituting and executing the differentiation, we obtain —

$$A = M \frac{\dfrac{ar}{C_2} + \dfrac{r^2}{B_3} + \dfrac{hr}{C_3}}{\dfrac{r(a+h)^2}{2C_3'} + \dfrac{ar^2}{C_2} + \dfrac{a^3}{12B_2} + \dfrac{2r^3}{3B_3} + \dfrac{hr^2}{2C_3} + \dfrac{\gamma a}{F_2G} + \dfrac{2\gamma r}{F_3'G} + \dfrac{\gamma h}{F_3G}}$$

And remembering that $M = Ak$, we see that the above result is identical with [10a].

37 For the determination of δ we have the equation —

$$\frac{dV}{dM} = \frac{d(V_1 + V_2 + \ . \ . \ . \ V_8)}{dM} = \delta$$

But remembering that now A is a function of M we have —

$$\frac{dV}{dM} = \frac{\partial V}{\partial M} + \frac{\partial V}{\partial A}\frac{dA}{\partial M} = \delta$$

in which $\dfrac{\partial V}{\partial M}$ is the partial derivative of V in respect to M. Since, however, $\dfrac{\partial V}{\partial A} = 0$ we have —

$$\frac{dV}{dM} = \frac{\partial V}{\partial M} = \delta$$

Hence δ is equal to the partial derivative of V in respect to M.

38 Executing the differentiation, we have —

$$\frac{2(M - Ar)a}{2C_2} + \frac{2Mr - Ar^2}{B_3} + \frac{2\left(M - \dfrac{Ar}{2}\right)h}{C_3} + \frac{2Mb}{C_1} = \delta$$

and with $A = \dfrac{M}{k}$, this equation becomes identical with [7d].

39 In another paper the author proposes to discuss the two- and three-cylinder two-bearing crankshaft, the multi-throw crankshaft, and the effect of clearances in the bearings.

No. 1865

THE DESIGN OF COOLING TOWERS[1]

By C. S. Robinson, Cambridge, Mass.[2]
Non-Member

The author points out that engineers have designed cooling towers in the past on empirical information and in accordance with the experience of previous successful designs. Wide departure from standard designs is difficult because of the lack of scientific basis for the design. The author therefore establishes the general principle applicable to cooling-tower design and derives equations for the use of the designer. He presents a quantity of experimental data to substantiate the validity of his formulas and shows by an actual experiment how these formulas are applicable to the design of a counter-current cooling tower.

THE factors influencing the design of cooling towers have been studied by a number of investigators. There has been, however, no statement of the influence of these factors of such a nature that it has been possible to calculate easily the performance of a given tower under widely varying internal and external conditions. Engineers have found that towers of specified construction, when operated at specified air and water rates, may be expected to cool water to within a certain number of degrees of entering air temperature and to discharge the air from the tower nearly saturated at some temperature approaching that of the entering water. In the absence of anything but empirical information, towers are therefore constructed along certain well-established and successful lines. Thus the designer of the forced-draft tower allows the water to fall through a height of perhaps 22 ft. and proportions the ground area on the basis of $6\frac{1}{2}$ gal. per sq. ft. of ground area per min. However, this empirical knowledge will not enable the engineer to predict what will happen when conditions depart widely from standard practice, as, for instance, the capacity of a tower 11 ft. or 44 ft. high. Nor could he estimate the tower height required to cool a given volume of

[1] This paper is based upon the principles demonstrated by Prof. W. K. Lewis, see paper No. 1849 of this volume of Trans., and is an application of these principles to one particular field.

[2] Department of Chemical Engineering, Massachusetts Institute of Technology.

Presented at the Annual Meeting, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

air from Death Valley at 120 deg. fahr. and 4 per cent relative humidity to 70 deg. fahr. by means of a parallel-current tower.

2 Investigations carried on by the Department of Chemical Engineering at the Massachusetts Institute of Technology under the direction of W. K. Lewis in connection with humidification and air drying, have led to the development of fundamental conceptions as to the mechanism involved in the transfer of heat between liquids and gases and in the vaporization and condensation of liquids and vapors. It is the purpose of this paper to show how these concepts can be applied to the particular case of cooling towers, and to devise by these means methods by which the engineer can simplify their design.

GENERAL PRINCIPLES

3 There are two principles upon which all of the subsequent work will be based, (*a*) the conservation of matter and energy, and (*b*) the potential concept. The latter may be expressed briefly as the effect upon the rate of flow of matter or energy of the driving force applied.

4 *The conservation of energy* as applied to a cooling tower may best be shown by means of a heat balance. This can be written as —

$$s_w w(t_1 - t_0) = Ws'(T_1 - T_0) + Wr'(H_1 - H_0) \quad . \quad . \quad . \quad [1]$$

where s_w = average specific heat of the water between the bottom and top of the tower

w = weight of water leaving the tower

s' = humid heat[1] of the air entering the tower, that is, the heat required to raise one degree in temperature one pound of dry air plus the water vapor H that it contains

T = temperature of air

t = temperature of water

W = weight of air (moisture free) entering tower

r' = total heat of water vapor at the top of the tower at the temperature of the leaving air minus the heat of the liquid of water at the temperature of the entering water.

The subscripts 1 and 0 refer to the top and bottom of the tower, respectively.

[1] Wm. S. Grosvenor, Trans. Am. Inst. Chem. Engrs., 1908.

5 Equation [1] is used universally by engineers at the present time in cooling-tower problems, usually in the approximate form —

$$w(t_1 - t_0) = Ws(T_1 - T_0) + Wr(H_1 - H_0) \quad . \quad . \quad . \quad . \quad [2]$$

where s and r are the average humid heat and latent heat between the top and bottom of the tower, respectively.

6 *The potential concept* may be applied to both the rate of transfer of heat from the liquid to the gas and to the rate of diffusion of water vapor from the liquid to the gas, as shown in the following paragraphs.

7 The rate or flow of heat is proportional to the temperature difference between the liquid and the gas, and the rate of diffusion of water vapor is proportional to the difference between the vapor pressure of the liquid water and the partial pressure of the water vapor in the gas.

8 The amount of heat flowing from the water to the gas per unit time would therefore be —

$$WsdT = haAdx(T - t) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where h is the coefficient of heat transfer per unit area (sq. ft.), a the square feet of cooling surface per cubic foot of volume of the tower, A the cross-sectional area of the tower, and x the height of the tower.

9 In the same way the weight of vapor vaporizing per unit time would be —

$$wdH = k'aAdx(P' - p) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where k' is the diffusion coefficient in pounds per unit area of exposed surface, P' the vapor pressure of the water, and p the partial pressure of the water vapor in the air. Since for small partial pressures p is nearly proportional to the absolute humidity H, Equation [4] may be rewritten as —

$$WdH = kaAdx(P - H) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

where P, the vapor pressure of the water, is expressed in terms of absolute humidity, that is, the absolute humidity of saturated air at the temperature of the water in question.

10 Dividing [5] by [3] gives —

$$\frac{dH}{sdT} = \frac{k(P - H)}{h(T - t)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

11 The mechanism by which the heat passes from water to the air may be understood by considering that the heat flows first

from the interior of the water to the surface, and then from the surface through a substantially stationary air film in contact with the surface to the moving air.

12 It has been shown[1] that $h/k = s$ when h refers to the coefficient of heat transfer through the air film only. If the total coefficient of heat transfer from the interior of the water to the moving air be used instead of h, the ratio h/k would not equal s, but would only approximate it to a greater or lesser degree according to whether the heat flow through the water took place easily or with difficulty as compared with the flow through the air film.

13 Unless otherwise noted h and k will henceforth refer to the overall coefficient of heat transfer and vapor diffusion respectively.

14 Therefore, taking $h/k = s$, [6] becomes —

$$\frac{dH}{dT} = \frac{P - H}{T - t} \quad \text{[7]}$$

or —

$$\frac{dH}{P - H} = \frac{dT}{T - t} \quad \text{[8]}$$

Equation [7], stated in words, says that the *differential* increase in humidity of the air is to the *differential* increase in temperature of the air as the humidity difference between the air and the water is to the temperature difference between the air and the water.

15 Equation [7] or [8] when integrated between proper limits would give the change in humidity and temperature of the water in its passage through the cooling tower. However, while the humidity of the air and the vapor pressure of the water are related to each other and the temperatures of the air and water are also related, the relationships are nevertheless of such a nature that exact integration is impossible, and an approximation is obtained by the following assumption.

16 It will be noted that the temperature difference between the water and the air in a counter-current cooling tower does not change greatly between the top and bottom of the tower, nor does the difference in humidities between the water and air. It is therefore a justifiable assumption that the average temperature difference is approximately equal to the arithmetical mean temperature difference, and that the average humidity difference is approximately equal to the arithmetical mean humidity difference. It may therefore

[1] W. K. Lewis, The Evaporation of a Liquid into a Gas, paper No. 1849 of this volume of Trans.

be said that for the whole tower, the total increase in humidity of the air is to the total increase in temperature of the air as the arithmetical mean humidity difference between the air and the water is to the arithmetical mean temperature difference between the air and the water.

17 For the tower shown in Fig. 1 where the subscripts 0 and 1

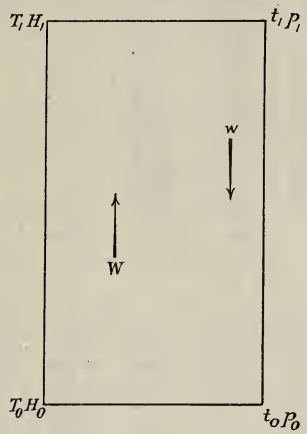


FIG. 1 DIAGRAMMATIC REPRESENTATION OF COUNTER-CURRENT COOLING TOWER

represent conditions at the bottom and top, respectively, there may now be written —

$$\frac{H_1 - H_0}{T_1 - T_0} = \frac{H_1 - P_1 + H_0 - P_0}{T_1 - t_1 + T_0 - t_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

which is the integrated form of Equation [7].

EXPERIMENTAL DATA

18 The author has collected and arranged in Table 1 the results of twenty-three tests made by himself and others on various types of cooling towers.

19 Tests 1 to 5 were on a slat type of forced-draft tower, 6 and 7 were on the same tower operating under natural draft, and Test 23 was on a Wheeler-Balke tower, all described in the Journal of the Ohio Society of Mechanical, Electrical and Steam Engineers, vol. 7.

20 Tests 8 and 9 were on a forced-draft slat-type tower described in the Journal of the American Society of Refrigerating Engineers, vol. 3, 1916–17, p. 32.

TABLE 1 COOLING-TOWER TEST DATA

Test No.	Tower No.	T	T_1	t_1	t_0	Rel. hum.	Gal. per min.	Q	V
1	1	71	90	105	84.7	40	651	110,500	53,900
2	1	72	93	107.8	87.5	60	638	108,000	50,100
3	1	64	96	112	88.5	60	638	124,500	51,400
4	1	69	92	108.5	87	48	643	115,000	52,200
5	1	83	95	109.9	90.5	48	640	103,400	50,600
6	1	43	101	116	98	75	632	94,800	23,500
7	1	60	118	135	115.8	73	630	102,000	15,575
8	2	82	99	115	83	65	617[c]	176,800	72,700
9	2	75	96	109	81	74	668	168,800	72,700
10	3	25	80[a]	87.5	71.3	72	590	78,000	.....
11	4	75.8	87.7	95.1	89.1	85	248	12,400	11,100[c]
12	4	77.0	103.9	112.1	99.2	78	230	24,700	9,060[c]
13	4	79.0	112.1	123.3	105.5	76	219	32,400	8,490[c]
14	4	76.1	89.9	94.4	87.9	60	288	15,500	9,700[c]
15	4	76.9	101.7	106.5	95.5	60	278	25,400	8,930[c]
16	4	77.0	112.9	123.7	105.2	57	259	39,800	8,970[c]
17	4	72.1	91.2	94.5	88.0	78	254	13,700	8,090[c]
18	4	73.0	105.1	113.7	99.9	75	246	28,100	9,030[c]
19	4	76.6	115.0	132.1	109.4	68	237	44,700	9,590[c]
20	4	72.9	90.9	92.4	88.4	87	269	8,900	6,160[c]
21	4	72.3	105.4	113.3	100.6	89	257	27,000	8,870[c]
22	4	72.7	115.2	128.2	108.1	88	241	40,200	8,800[c]
23	5	91	106	109	97	59	3,200	319,000	125,000[c]

Test No.	Tower No.	H_0	P_0	H_1	P_1	H	W	Δw	$ka \times$ vol.
1	1	0.006	0.026	0.031	0.050	0.0195	3700	93	4,800
2	1	0.010	0.028	0.034	0.055	0.0195	3410	82	4,200
3	1	0.008	0.029	0.037	0.062	0.023	3460	100	4,300
4	1	0.007	0.028	0.033	0.056	0.022	3560	89	4,000
5	1	0.012	0.031	0.036	0.058	0.0205	3420	82	4,000
6	1	0.004	0.040	0.044	0.071	0.0315	1550	62	2,000
7	1	0.008	0.070	0.075	0.129	0.057	915	61	1,100
8	2	0.015	0.024	0.041	0.068	0.018	5200[c]	135	7,500
9	2	0.014	0.023	0.037	0.057	0.0145	5276[c]	121	8,300
10	3	0.002	0.016	0.022	0.028	0.010	5200[c]	99.5	.....
11	4	0.015	0.030	0.027	0.036	0.012	790[c]	9.5	790
12	4	0.015	0.041	0.045	0.063	0.022	648[c]	19.4	880
13	4	0.016	0.051	0.059	0.090	0.033	606[c]	26.7	810
14	4	0.011	0.029	0.029	0.035	0.012	693[c]	12.5	1,040
15	4	0.011	0.037	0.043	0.052	0.018	638[c]	20.4	1,130
16	4	0.011	0.050	0.061	0.090	0.034	641[c]	32.0	940
17	4	0.013	0.029	0.031	0.036	0.0105	578[c]	10.4	990
18	4	0.013	0.042	0.047	0.066	0.024	644[c]	21.9	910
19	4	0.013	0.058	0.065	0.118	0.049	692[c]	36.0	730
20	4	0.015	0.029	0.030	0.033	0.0085	440[c]	6.6	780
21	4	0.015	0.043	0.048	0.065	0.023	633[c]	20.9	910
22	4	0.015	0.055	0.066	0.104	0.039	628[c]	32.0	820
23	5	0.019	0.039	0.051	0.057	0.013	8250[c]	273.0	21,000

Test No.	Tower No.	Q_{sens}	Δt	$ha \times$ vol.	h/k	ha	ka	u	$100 \times \frac{ka}{u}$
1	1	18,000	14.4	1250	0.26	0.57	2.18	540	0.40
2	1	21,000	15.2	1380	0.33	0.63	1.91	500	0.38
3	1	32,000	20.3	1570	0.36	0.71	1.96	510	0.38
4	1	24,000	17.3	1390	0.35	0.63	1.82	520	0.35
5	1	12,000	10.7	1120	0.28	0.51	1.82	510	0.36
6	1	26,000	35	740	0.37	0.34	0.91	240	0.38
7	1	15,000	36.4	410	0.37	0.19	0.50	160	0.31
8	2	23,000	24.5	1470	0.20	0.46	2.36	320	0.77
9	2	29,000	25	1720	0.21	0.54	2.61	320	0.82
10	3	.....	....	....	....	....	....	...	....
11	4	2,400	10.4	230	0.29	0.17	0.57	174	0.33
12	4	4,300	15.2	283	0.32	0.20	0.64	142	0.45
13	4	4,400	18.9	233	0.29	0.17	0.59	133	0.44
14	4	2,400	8.2	292	0.28	0.21	0.75	151	0.50
15	4	4,000	11.7	342	0.30	0.25	0.82	140	0.58
16	4	6,200	19.5	318	0.34	0.23	0.68	140	0.49
17	4	2,800	9.6	292	0.30	0.21	0.72	126	0.57
18	4	5,100	18.9	270	0.30	0.21	0.66	141	0.47
19	4	6,900	25.0	276	0.38	0.20	0.53	150	0.35
20	4	2,000	8.4	238	0.31	0.17	0.57	96	0.59
21	4	5,000	18.1	276	0.30	0.20	0.66	139	0.47
22	4	6,600	24.2	273	0.33	0.20	0.59	125	0.47
23	5	31,000	4.5	7,000	0.33	....	....	...	....

[a] Assumed values. [c] Calculated values.

21 Test 10 was on a natural-draft tower described in the Transactions of The American Society of Mechanical Engineers, vol. 31, 1909, p. 75.

22 Tests 11 to 22 were on an experimental Badger spray-type tower erected at the Massachusetts Institute of Technology, and the results have not been previously published.

23 The method of calculating the data may be explained by reference to Test 1.

T_0 = temperature of inlet air, deg. fahr.
T_1 = temperature of outlet air, deg. fahr.
t_1 = temperature of inlet water, deg. fahr.
t_0 = temperature of outlet water, deg. fahr.
Rel. Hum. = relative humidity in per cent
Gal. Min. = gallons of water circulated per minute
Q = heat given up by water in B.t.u. per minute
$= 651 \times 8.3 \times (105 - 84.7) = 110{,}500$
V = cubic feet of air per minute
H_0 = absolute humidity of inlet air in pounds of water per pound of dry air, as read from humidity chart [1]
P_0 = equivalent absolute humidity of outlet water, i.e., the absolute humidity of air saturated at the water temperature
H_1 = absolute humidity of outlet air, assuming saturation
P_1 = equivalent absolute humidity of inlet water
H = mean humidity difference between the air and water

$$= \frac{0.026 - 0.006 + 0.050 - 0.031}{2} = 0.0195$$

W = pounds of dry air per minute, as calculated from the volume of air V divided by the humid volume (the cubic feet occupied by one pound of dry air plus the water vapor which it contains, as read from the humidity chart)
w = pounds of water evaporated per minute
$w = W(H_1 - H_0) = 3700(0.031 - 0.006) = 93$
$ka \times$ vol. = pounds of water evaporated in the whole tower per minute per one pound mean humidity difference

$$= \frac{93}{0.0195} = 4800$$

[1] Wm. S. Grosvenor, Trans. Am. Inst. Chem. Engrs., 1908.

Q_{sen} = sensible heat picked up by the air per minute

= $Ws(T_1 - T_0)$, where s is the humid heat as read from the humidity chart[1] ($s = 0.25$)

$= 3700 \times 0.25(90 - 71) = 18{,}000$ B.t.u.

t = mean temperature difference between air and water

$= (105 - 90 + 84.7 - 71) / 2 = 14.4$

$ha \times$ vol. = B.t.u. picked up as sensible heat by air in the whole tower per degree temperature difference

$$= \frac{18{,}000}{14.4} = 1250$$

$$\frac{h}{k} = \frac{ha \times \text{vol.}}{ka \times \text{vol.}} = \frac{1250}{4800} = 0.26$$

$$ha = \frac{ha \times \text{vol.}}{\text{vol.}}$$ where "vol." is the volume of the tower.

24 The dimensions of the tower in Test 1 were not given in the article, but other information led to the assumption that the tower was $10 \times 10 \times 22$ ft., a volume of 2200 cu. ft., whence —

$$ha = \frac{1250}{2200} = 0.57 \text{ and } ka = \frac{4800}{2200} = 2.18$$

u = linear velocity of air through the total cross-section of the tower

$$= \frac{53{,}900}{10 \times 10} = 540 \text{ ft. per. min.}$$

$$\frac{ka}{u} = \frac{2.18}{540} = 0.0040$$

25 The most interesting tests are those numbered 1 to 7, which have been quoted several times in articles by different investigators. In these tests the first five were under forced draft, while the last two were under natural draft, with much reduced air velocity. A plot of ka versus air velocity for these seven tests gives, as shown in Fig. 2, a straight line passing through the origin.

26 It has been noticed by others [2], that the rate of cooling in towers is approximately proportional to the air velocity. Investigations at the Institute being carried on at the present time indicate that ka for humidifying apparatus in general is probably proportional to a power function of the air velocity something less than one, but

[1] Wm. S. Grosvenor, Trans. Am. Inst. Chem. Engrs., 1908.

[2] Jour., Am. Soc. Refrig. Engrs., vol. 3, 1916–17, p. 32.

for the present, at least, linear proportionality is a fairly close approximation. While k is the actual coefficient of diffusion per square foot of surface, since in many types of towers the active surface cannot be accurately measured the coefficient ka is used instead, where a represents the actual surface in square feet per cubic foot of tower. In most towers a is uniform throughout the volume of the tower.

27 It is therefore justifiable to call the value ka/u the "tower constant" and to use the value of this constant as a means for comparing the operation of towers of various sizes and types. Thus for tower No. 1 the average value of ka/u is 0.0037 and the average deviation from this value is 0.00021, or less than 6 per cent, while the maximum deviation is 0.0006, or 16 per cent. The important

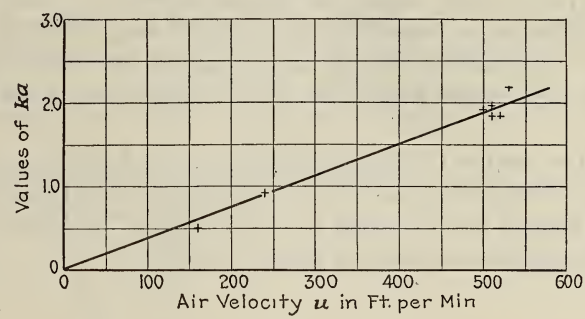


Fig. 2 Relation Between ka and Air Velocity

thing indicated by these tests is that ka/u was unaffected whether the tower was operated with forced or with natural draft.

28 Tower No. 4 has an average value of 0.00475 for ka/u, with an average deviation of 10 per cent and a maximum deviation of 31 per cent.

29 Tower No. 2 has an average constant of 0.0080, which is high compared with the other towers tested. The drop in pressure through this tower was not published. It would be interesting to compare its drop in pressure with that in other towers which show smaller tower constants. It is of course obvious that a high tower constant can be obtained by obstructing the flow of air through the tower by cutting down the mean free area, but this again is at the expense of friction and back pressure. Wherever possible the drop in pressure through the tower should be measured and published.

30 The average value of the humid heat of air in a cooling

tower is about 0.25. It will be noted that the column h/k in Table 1, which should be approximately equal[1] to s, has values of the same order of magnitude in general, but which are somewhat higher. The reason for this as predicted from the statements made in Par. 12 is shown by the fact that, while the surface of the water is at a lower temperature than the interior, making the true h for the air film greater, the value of k was calculated for the average water temperature instead of using the surface temperature. The latter is lower and would give a higher value for k. But since the vapor pressure of water rises more rapidly than the temperature, the ratio of h to k calculated on the average water temperature would be greater than that calculated on the lower surface temperature, which was the case in all but two cases in Table 1. The experimental data are therefore offered as proof of the validity of Equation [**7**].

31 Information regarding the size of tower in Test 23 was not available. It would be interesting to compare the values of ha and ka for this large tower with those of the smaller ones in the previous tests.

32 The results of the calculations of the test runs are felt to confirm in a remarkable manner the conceptions regarding humidification in general as developed by W. K. Lewis, furnishing as they do ample experimental proof of their validity.

APPLICATION OF THE EQUATIONS DEVELOPED TO TOWER DESIGN

33 The value of the foregoing equations with respect to cooling-tower design may best be shown by means of an example.

34 In any cooling tower the law of conservation of energy must apply. This law is represented by the heat balance of Equation [**1**]. Furthermore, in any cooling tower there is a necessary relationship between the amount of heat transferred by conduction and that eliminated by evaporation which is represented by Equation [**9**]. Finally, the capacity of any cooling tower is determined by the rate of transfer of heat and the rate of diffusion of vapor in it. These rates are dependent upon the design of the tower, and can only be determined on the basis of experimental data on the performance of towers of the type in question. This capacity factor is covered by Equation [**3**] or Equation [**5**] in the differential form, but in actual

[1] W. K. Lewis, The Evaporation of a Liquid into a Gas, paper No. 1849 of this volume of Trans.

design it is more satisfactory to use an integrated form of Equation [**5**] obtained by the use of arithmetical mean humidity differences and represented by Equation [**10**]. The combination of these three equations represents the conditions that must obtain in any tower and therefore serve as a proper basis for tower design.

35 The coefficient ka/u expresses the volumetric capacity of any particular type of tower. Experimental determination of this constant is necessary before design can be accomplished.

36 It is necessary for the engineer to select the type of tower most suitable for his purpose, and, from previous tests on towers of similar type, obtain values of ka which can be anticipated. In general certain specifications must be met. These may be:

Weight of water to be cooled, w
Temperature of water to be cooled, t_1
Temperature to which water must be cooled, t_0
Average (or worst) outside air temperature, T_0
Average (or worst) outside air humidity, H_0

for which the following values may be taken:

w = 3000 gal. per min.
t_1 = 110 deg. fahr.
t_0 = 80 deg. fahr.
T_0 = 80 deg. fahr.
H_0 = 0.0130 (60 per cent relative humidity)

those for T_0 and H_0 being the worst atmospheric conditions under which a tower must cool the water at 80 deg. fahr.

37 It will be noted that the condition of reducing the temperature of the water to that of the entering air is exceptionally severe and will call for a tower considerably larger than usual, since towers rarely have to meet such specifications.

38 There are in this case three unknown conditions, the temperature and humidity of the outgoing air, and either the volume of the tower or the ratio of the amount of air to the amount of water. The air-water ratio is usually determined by the type of tower selected and is therefore known, leaving the volume of the tower as the third unknown.

39 In order to calculate these unknown quantities three independent equations are needed. These may be taken as Equation [**2**],

$$w(t_1 - t_0) = Ws(T_1 - T_0) + Wr(H_1 - H_0)$$

Equation [9],

$$\frac{H_1 - H_0}{T_1 - T_0} = \frac{H_1 - P_1 + H_0 - P_0}{T_1 - t_1 + T_0 - t_0}$$

and an integrated form of Equation [4], obtained by employing the same assumption as that used in integrating Equation [7],

$$W(H_1 - H_0) = kaAx \frac{P_1 + P_0 - H_1 - H_0}{2} \quad . \quad . \quad . \quad . \quad [10]$$

Equations [2], [9], and [10] may be solved simultaneously for H_1, T_1, and Ax (which latter equals the volume of the tower). The author has seen fit to solve for W instead of Ax, but after H_1 and T_1 have been found it is easy to convert the solution for W into that for Ax. The final values are as follows:

$$T_1 = -T_0 + t_1 + t_0 - \frac{\dfrac{2w(t_1 - t_0)}{kaAx}}{s + r\dfrac{2H_0 - P_1 - P_0}{2T_0 - t_1 - t_0}} \quad . \quad . \quad . \quad . \quad . \quad [11]$$

$$H_1 = -H_0 + P_1 + P_0 - \frac{\dfrac{2w(t_1 - t_0)}{kaAx}}{s\dfrac{2T_0 - t_1 - t_0}{2H_0 - P_1 - P_0} + r} \quad . \quad . \quad . \quad . \quad [12]$$

$$W = -\frac{\dfrac{w(t_1 - t_0)}{s(2T_0 - t_1 - t_0) + r(2H_0 - P_1 - P_0)}}{1 + \dfrac{2w(t_1 - t_0)}{kaAxs(2T_0 - t_1 - t_0) + r(2H_0 - P_1 - P_0)}} \quad . \quad [13]$$

These equations become much simplified when the known numerical values are substitutes for the literal quantities.

40 In the particular problem in question, suppose that a forced-draft slat-type tower such as was used in Tests 1 to 7, whose tower constant ka/u is 0.0037, be selected. Towers of this type are found to be economical when handling 6.5 gal. of water per min. per sq. ft. of ground area with an air velocity of about 500 ft. per min.

41 The area of the proposed tower will therefore be 3000/6.5 = 460 sq. ft., and the volume of air will be 500 × 460 = 230,000 cu. ft. per min. The humid volume of one pound of dry air as read from the humidity chart is 13.8 cu. ft. Therefore —

$$W = \frac{230{,}000}{13.8} = 16{,}700 \text{ lb. per min.}$$

also —

$w = 3000 \times 8.3 = 25{,}000$	$r = 1050$ (approx.)
$t_1 = 110$	$H_0 = 0.013$
$t_0 = 80$	$P_1 = 0.0585$
$s = 0.24$	$P_0 = 0.022$
$T_0 = 80$	$ka = 0.0037 \times 500 = 1.85$

Substituting these values in Equation [**13**] and solving for Ax gives 41,500 cu. ft. for the volume of the tower, and since the ground area was determined to be 460 sq. ft., the tower height would be 41,500/460 = 90 ft. The severe conditions imposed account for the great height required.

42 If the tower be built as calculated, the performance under any other atmospheric conditions may be readily calculated by substitution in the proper equation.

43 The author realizes that the foregoing methods of calculating cooling-tower performance do not form the complete solution of the problem, and that the ultimate design of the best tower will depend upon the striking of an economic balance between the size of the tower and the cost of moving the water and the air. He feels, however, that this method of calculation is distinctly in advance of anything which has thus far appeared in print, that it furnishes a convenient and accurate tool for the designing engineer, and that it is the necessary basis for the economic balance referred to.

44 Finally, the author wishes to urge the inclusion of more complete and more accurate data in published accounts of tower performances. Of all the published tests studied, only those tabulated had sufficient data to enable them to be analyzed, and even then, in most cases, assumptions were necessary. Inaccurate data are often common. Test No. 10 is an example of this, the heat of vaporization alone being considerably greater than the total cooling of the water, which inaccuracy renders the test useless for purposes of analysis. A more complete knowledge of the effect of varying conditions on *ha* and *ka* can only come from studies of large numbers of accurate tests, and it is upon such increased knowledge that advance in tower design depends.

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DISCUSSION

B. H. Coffey. The members of the American Society of Refrigerating Engineers present will remember that Mr. George Horn and the writer have devoted much time and study to this subject. We are therefore gratified to see the theory of cooling towers becoming of interest to the scientific men of college faculties as instanced by the present paper. If the unequalled experimental facilities and mathematical ability of our great engineering schools become engaged upon this subject, we believe it can be shortly put upon a practical basis for the general profession.

All cooling towers are mere structural assemblages of cooling surface, the cost of which is by far the largest item in the installation. For this reason the designer and purchaser are peculiarly interested in the area of surface required to do the specified cooling. The discussion will therefore be confined to the points bearing on this part of the subject.

Equations [3] and [4] show the dual heat currents that always exist between air and water when not in thermal equilibrium, both expressions containing a cooling surface term and transmission coefficient. In Equation [3] the potential or driving force is temperature difference and in Equation [4] pressure difference. For future reference we wish here to point out that by simple transposition in each equation the surface increment equals the heat increment multiplied by the reciprocal of the potential and consequently for zero potential the surface increment is infinite.

By ingenious use of the connecting constant s, Equation [8] is established showing the relation between the latent and sensible heat currents. The author fails to integrate this equation and resorts to an approximation with which we take issue.

The approximation (Par. 16) is based upon the assumption that the mean of the extreme differences is the true mean and the assumption is based upon the stated fact that in counter-current cooling towers the pressure and temperature differences do not change greatly between the top and bottom of the tower. No evidence is offered to support this statement which we dispute. Referring to Par. 20, Table 1, Tests 1 to 7, the temperature differences at the bottom of the tower as a percentage of those at the top range from 91 to 366 per cent and on the same basis the pressure differences vary from 83 to 148 per cent. We think these figures are sufficient to cast grave doubts upon the author's statement

and basis of his approximation and that probably the mean potentials he obtains are not the true means.

If the mean potentials are incorrect the mean transmission coefficients h and k which are derived from them (Par. 23) are incorrect and the ratio h/k will be incorrect. Referring to Table 1, col. h/k, we find this ratio varies from 0.20 to 0.38 when it should be approximately constant at 0.25 (Par. 30). The explanation of these wide discrepancies given in this paragraph are to us not convincing.

The extraordinary height of tower 90 ft. (Par. 41) to meet the not particularly severe conditions of Par. 36 as obtained from the simultaneous Equation [**13**], Par. 39, is, we believe, due to incorrect mean potentials.

As a further test of Equation [**13**] we used for final water temperature 69.7 deg. the wet-bulb temperature of the entering air. Under these conditions the total potential becomes zero and the cooling surface infinity as pointed out above. The calculation resulted in a tower 230 ft. high instead of infinity which we again attribute to incorrect mean potentials.

In our opinion this line of attack upon the cooling tower problem while promising has failed to produce working formulas that express the cooling process. We suggest as a basis using total potential equations. The relation $h/k = s$ gives a means of converting temperature potential into equivalent pressure potential or the reverse. The total potential can be thus expressed either in temperature or pressure alone and the problem much simplified.

We note with regret the absence of any reference to the wet-bulb temperature in this paper, a physical quantity universally regarded of the greatest importance in this subject.

W. M. Grosvenor.[1] It is a real satisfaction to find after fourteen years that a piece of one's own work stands the test of time and is still of use to other engineers, particularly in such a very admirable consideration of cooling towers as the author has given us. The article on Calculations for Dryer Design to which he refers was published in the *Proceedings* of the American Institute of Chemical Engineers and the *Heating and Ventilating Magazine* for 1908 when the best available data on humidity were those of the U. S. Weather Bureau and on these figures the calculations were based and the resulting curves plotted. The conception was

[1] Cons. Chemist and Factory Engr., 50 E. 41 Street, New York, N. Y.

there introduced of representing what might be called adiabatic evaporative cooling by lines intersecting the curves of relative humidity on a chart having temperatures as one ordinate and weight of moisture per lb. of air, volume per lb., B.t.u. per lb. of air when damp (humid heat), etc., on the other ordinate. This has proven a very useful and easy way of calculation. Some three years later at the Annual Meeting of The American Society of Mechanical Engineers for 1911 in a paper entitled Rational Psychrometric Formulae, Willis H. Carrier used this method of presenting a newly calculated set of curves based, I believe, on more accurate data than those of the Weather Bureau. It seems to the writer that he did a very excellent and valuable piece of research work that was very thoroughly discussed at that meeting but has received too little attention since. This valuable paper contains no reference to any previous publication of humidity charts with adiabatic cooling lines, etc., but the writer wants to call Mr. Robinson's attention to Mr. Carrier's paper and to ask for it the careful consideration it apparently deserves.

Now that we have in the author's work the foundation laid for a more logical and clear understanding of the data needed for perfecting the design of cooling towers, it becomes the obvious duty of engineers having cooling towers under operation to determine as well as they can the conditions of operation and communicate the information to him. He is then in position to suggest changes in operating conditions if not in design and, on the basis of the results before and after, can revise and perfect what should be a very valuable solution of a problem that is becoming increasingly important with municipal growth.

The Author is glad of an opportunity to clear up an obscure point in his paper, as indicated by Mr. Coffey's discussion.

It is correct, as pointed out by Mr. Coffey, that the differences between the temperatures of the air and the water at the top and at the bottom of the tower in Tests 1 to 7 vary by as much as four fold in the extreme case. The statement of the author that the temperature differences remained substantially constant therefore seems far from the fact. However, the author had in mind when he made the statement, the fact that the integral of the Equation [8] requires the true mean temperatures and humidity differences. Not knowing the true means, and realizing that the differences varied not over four fold, it seemed perfectly proper to

assume the arithmetical mean temperature differences, since they would probably be not very far from the true mean. He based this statement on the fact that the log mean temperature difference would give a result only differing 16 per cent from the arithmetical mean difference when the difference is four fold, while the geometric mean difference would differ by less than 20 per cent, when the difference in the temperatures themselves varied by over 366 per cent. This does not mean that either of these two means is correct in this case, but it does show that in estimating mean values where the values themselves do not vary by more than four fold, the arithmetical mean is likely to be well within the experimental error of the tests.

The author therefore believes that within the limits of commercial requirements, his solution of the equation is practicable and satisfactory, and he does not believe that the error introduced by this approximation is the chief cause of the variation in the values of h/k obtained. The use of the arithmetical mean has the obvious effect, as pointed out by Mr. Coffey, of giving a finite answer when an infinite one is required by the theory, but since an infinite tower is beyond the range of actual practice, he still feels that the equations developed go at least one step further towards the solution of the cooling tower problem than anything that has heretofore appeared in print.

No. 1866

ASH HANDLING

By John Hunter and Alfred Cotton, St. Louis, Mo.
Members of the Society

Various methods of handling ash are described and illustrated, beginning with early developments on shipboard. The greater part of the paper, however, deals with stationary practice, beginning with rudimentary and progressing to the most modern installations, of which schematic and actual examples are given.

Methods of removing ashes from basement boiler rooms are followed by a discussion of the design, construction, and capacity of hopper ashpits, including their doors and water seals. Systems of mechanical conveyance and elevation are described, comprising ash cars, skip hoists and bucket conveyors. Fluid conveyance, as represented by water sluicing and steam-jet conveyors, is discussed in general, and typical examples of both are illustrated and described in detail. Particulars of ash bunkers and settling basins are followed by a discussion of the final disposal of ash.

ASH HANDLING is just as important as coal handling. It is a simpler problem only to the extent that there is less ash than coal, but the coal disappears while the ash accumulates. The ash must be got rid of or the plant will be put out of commission very quickly.

2 The first stage in ash handling is getting the ash out of the fire. This part of the subject does not concern us here, except that the methods employed in either hand or machine firing affect the design of ash-handling equipment, as also does the last stage, the final disposal. The first stage varies all the way from hauling the ashes out of the fire door by hand to washing them out of the gases. The final stage varies from wheeling them out in a wheelbarrow to carrying them away in scows and dumping them at sea. The ash-handling equipment must fit the conditions at both ends.

3 Ash handling was originally accomplished entirely by hand, but with the growth of the size of boiler plants it is now either

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partly or entirely mechanical. There are three general methods of conveyance in use: air, water, and purely mechanical.

4 The great development of the central electric generating station has compelled operating engineers to give much attention and thought to ash handling. It is only a few years since boilers of 600 hp. were considered large, while today 2000-hp. units are not at all uncommon and some 3000-hp. boilers are in use. Furthermore, while boilers were usually operated at about their rating of 10 sq. ft. of heating surface to the boiler horsepower, they are now commonly operated at twice their nominal rating; and in the larger central stations it is common practice to run them for short intervals at three or even four times their rating. The bearing of this development on ash handling may be strikingly seen when we remember that a 250-hp. boiler running at rating and burning coal with 15 per cent of ash would make 150 lb. of ash per hour; while each 2000-hp. unit at 200 per cent of rating makes well over a ton of ash per hour.

5 The subject may well be divided into three phases: stationary, locomotive, and marine. The locomotive phase may be passed over because of the ability of the locomotive to generate ashes over a wide range of country and dump them in one spot. Marine practice will be dealt with first, partly because it happens that the authors each had early experience in this line, and partly because definite progress in ash handling was first made on shipboard.

MARINE PRACTICE

6 Practically all coal-fired ships are hand-fired and hand-cleaned. No mechanical stoker has yet met the conditions sufficiently to warrant even an extended trial. The old way was to load the ash in buckets which were lifted through the ventilators by a steam "ash hoist" on deck. The noise was objectionable on passenger boats, as also was the dust flying all over the ship when dumping overboard.

7 About thirty years ago the engineers of one of the leading transatlantic steamship companies adopted the ash ejector. This was a simple arrangement in which a hopper provided with a water-tight lid was placed on the boiler-room floor, with a pipe leading from the hopper upward and through the ship's side above the water line. The ashes were dumped into the hopper and the lid

closed and clamped. Then water under as much as 200 lb. pressure was admitted to the hopper through a tapered nozzle, discharging the water and ashes overboard by a siphon action. This method was a vast improvement over the old ash hoist, and was much easier for the men, who would handle the ashes in half the time. Ash ejectors have been much improved and are now extensively used in passenger ships. Fig. 1 illustrates one of these ash ejectors as made by the M. T. Davidson Company, of New York.

8 In passenger ships it is often objectionable to discharge ashes at high velocity from the ship's side above the water line, as this frequently results in ashes flying over the deck. It is not

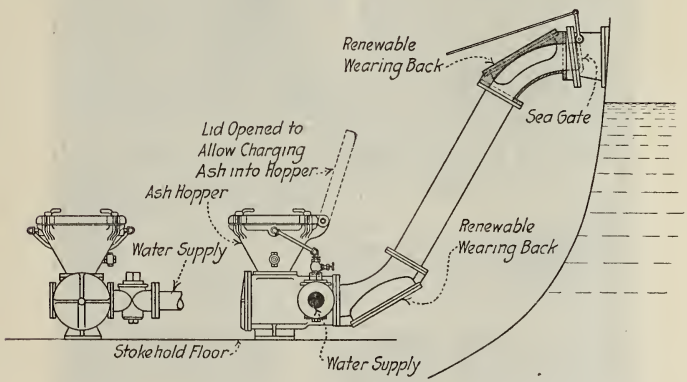


Fig. 1 Marine Ash Ejector, Operated by Water

permitted when in dock or when small boats are near. These difficulties are met by Stone's underline ash expeller which is illustrated in Fig. 2, whereby the ashes are discharged vertically downward through the ship's bottom. The method of operation is that of the water ejector having a water motive jet and an auxiliary water jet to drive the ashes from below the receiving grating into the ejector entrance. The passage from the ash-receiving grating to the ejector is closed by a valve attached to a piston operated by the pressure at the throat of the venturi portion of the ejector. When this valve is being closed, a supply of water is arranged to wash the valve and its seat clear of ash. Modified plug cocks are also provided for emergency closure in addition to the automatic valve.

9 Some of the incidental advantages of this system are interesting. Since the ashes are discharged thoroughly wet and well below the water line, they do not leave a spoor or trail of ashes on the surface which would convey information to an enemy in war time. Also, as each ejector can discharge 650 tons of water

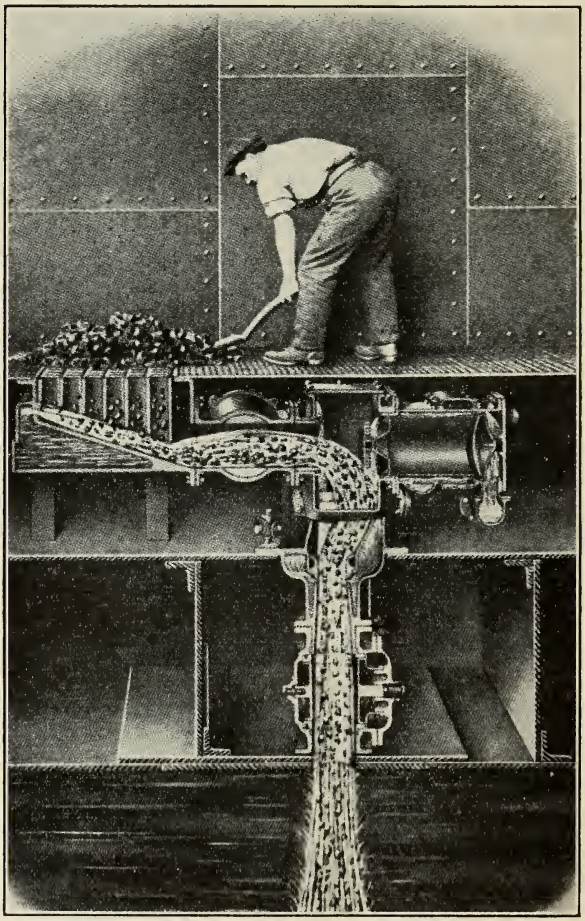


Fig. 2 Stone's Marine Ash Expeller, Operated by Water

per hour, they can be used as salvage pumps to overcome serious leakage. The British battleship *Marlborough* was torpedoed as she was going into action at Jutland, and enough of the ship's side was ripped off from the bilge to above the water line to expose two

watertight compartments. As the explosion sprung the bulkhead, between 300 and 400 tons of water per hour poured into the forward stokehold. Ordinarily such a leak would have meant the almost certain loss of the ship; but by using the ash expeller as a salvage pump the vessel kept her station and continued fighting throughout the battle. Not only that, but she was kept afloat by means of the ash expeller for six days until she could be docked. The authors are indebted to the McNabb Company, of Bridgeport, Conn., for the illustration forming Fig. 2, and for particulars of the incident just related.

10 The steam-jet ash conveyor is used to a considerable

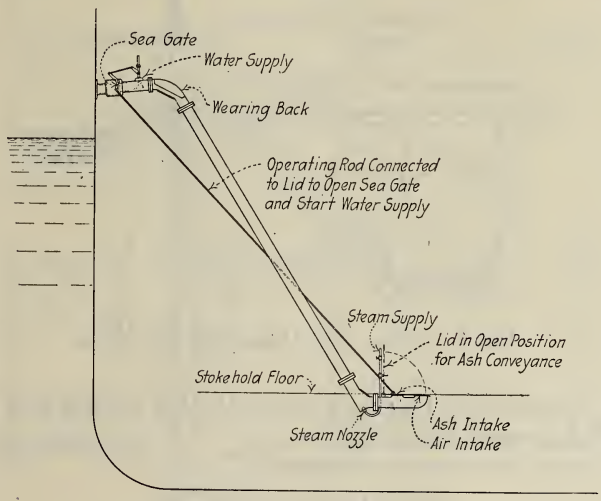


Fig. 3 Marine Ash Ejector, Operated by Steam

extent on harbor tugboats. Such an installation is that of the Victor Engineering Company, of Philadelphia, which is shown in Fig. 3. This type of conveyor will be more particularly described when dealing with stationary practice. The objection to its use in ocean-going steamers is the loss of water, since the steam used is discharged to the atmosphere. All water lost to the atmosphere by the open discharge of steam, such as from safety valves, whistle, leaks, etc., must be made up by the use of sea water; and this sea water must be purified either in the boilers by concentration and blow-down as was the old practice, or by the modern method of distillation. Therefore the amount of "make-up" required must

always be kept as low as possible, with the result that very few steam-consuming auxiliaries which do not exhaust into the condenser can find a place on ocean-going steamers. Harbor tugs carry fresh-water tanks which are easily and regularly replenished, and even their main engines are therefore frequently non-condensing. There is no objection to the use of steam-jet ash conveyors in these cases, provided their use is not forbidden by harbor rules. It will be understood that this objection has no reference whatever to the

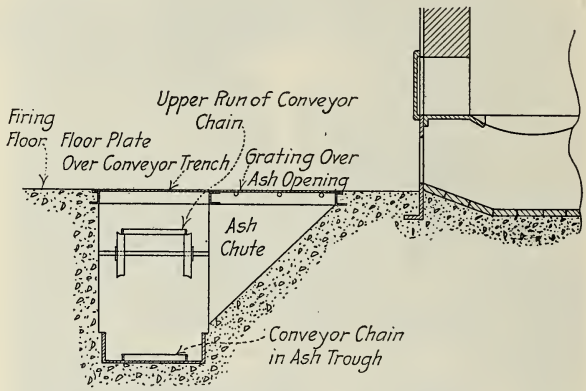


Fig. 4 Chain and Cross-Bar Conveyor

amount of steam used thermodynamically, but simply that it represents so much fresh water lost.

STATIONARY PRACTICE

11 Except that there are several standardized types of conveyors, classification of methods of ash handling is not convenient and would serve no useful purpose. Each problem must be considered entirely according to its particular circumstances. The methods of offering the ash in the first place, the general conditions to be met, and the final disposition are all so varied and the possible combinations of each are so numerous that each installation is different from others in some respects. Therefore no real classification will be attempted, but several different schemes and installations will be fully described. It is believed, after very mature consideration, that this will be the most satisfactory method of dealing with the subject, and will be most helpful to those con-

fronted with this problem. Component parts of installations illustrated may be taken and combined to meet most requirements; but it must be remembered that to make a successful installation in a large modern plant, considerable experience and shrewd judgment are very necessary.

12 The most rudimentary method is that in which the ashes are hoed out of the fire doors and ashpits on to the firing floor and shoveled into wheelbarrows which are dumped outside the boiler room, either to fill up depressions or to be carted away. In some cases the ashes are shoveled into industrial railway cars, which are then pushed to the dumping point. In some existing plants a

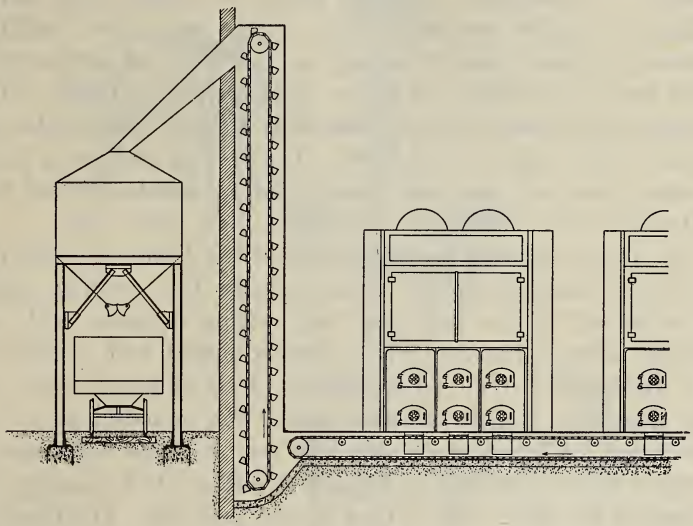


Fig. 5 Chain and Cross-Bar Conveyor with Bucket Elevator and Ash Bunker

basement is out of the question, and even if a change were made to stoker firing, the ashes would still have to be hoed out of the ashpit. But the wheelbarrow can be replaced by either a mechanical or air conveyor.

13 Fig. 4 is a cross-section of a block-chain conveyor running in a trench under the firing floor. The ashes are hoed on to the grating, through which they fall down the chute to the bottom of the trench. They are then drawn along by the chain, discharging into a bucket elevator as illustrated in Fig. 5, or on to an inclined chain conveyor carrying the ash into an elevated storage hopper.

14 Where there is a plentiful supply of water, a flume may be carried along under the firing floor and the ashes raked into it through openings. This is a very simple method, but it is seldom that a natural water supply and sufficient fall from the plant are available to adequately dispose of the ashes.

15 The problem is more complicated when the boiler room is considerably below the ground level. In such cases there will seldom be a basement below the boiler room, partly on account of the expense of further excavation, and often because of difficulties with building foundations and with water. Good examples are the boiler rooms of office buildings, hotels, etc., where the boilers are generally a considerable distance below the street level and the building is closely surrounded by streets and other buildings. Whatever the final disposition may be, the ashes are invariably removed from the vicinity in motor or horse-drawn trucks. Here, the important consideration is principally that of hoisting the ashes from the boiler room to the street. The crudest method is to load the ashes into ash cans and hoist them by hand-operated block and tackle to the street. This entails high labor cost.

16 In the isolated heating plants of the Union Electric Light and Power Company in St. Louis, exactly this problem existed. It was solved in the following very simple manner with some saving in labor: The ash trucks were equipped with a davit just like a regular boat davit on shipboard. A small winch with a 4-in. barrel geared to a $\frac{1}{4}$-hp. electric motor was mounted at the side of the truck. A $\frac{3}{16}$-in. steel wire rope was attached and wound on the winch barrel, and passed through a pulley block hanging from the head of the davit with a hook at the free end. The davit was simply a length of $1\frac{1}{2}$-in. pipe bent to shape with the head end flattened to facilitate bolting on the lug of the pulley block. Current for the motor was conveyed by a flexible cord from a socket in the plant and snapped to the motor when the ash truck arrived. Opening the sidewalk cover, the hook was dropped and the first ash can attached, quickly hoisted to the davit head, the davit swung around and the ash can dumped into the wagon, and then returned to the basement. With this cheap and simple rigging the truck was quickly loaded, the flexible cord disconnected from the motor, the sidewalk cover closed, and the truck driven away. The whole operation was completed in a few minutes with greatly increased economy in handling.

17 In some instances the air conveyor (the so-called steam-jet conveyor) has been used very advantageously in hospitals, hotels, and office buildings in crowded parts of cities, but it is not always applicable. Unless certain precautions are observed, such as connecting the air discharge into the chimney, the noise may entirely prevent the use of the system.

18 Another method is the " post box " elevator illustrated in

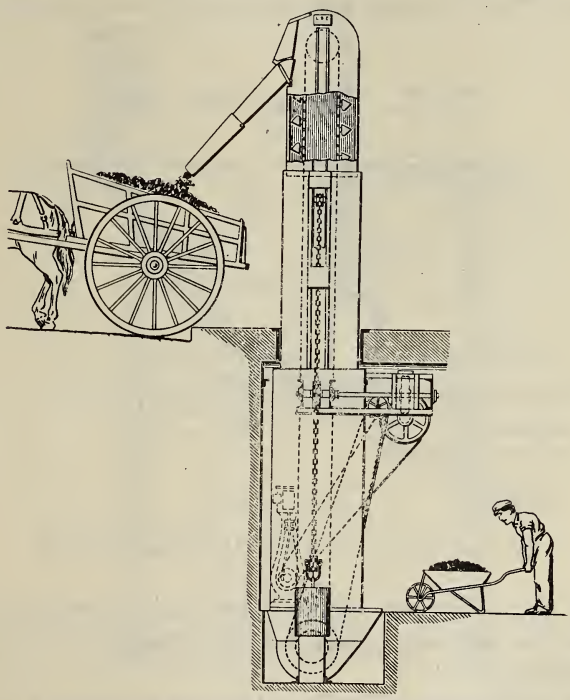


Fig. 6 Telescopic Sidewalk Bucket Elevator

Fig. 6. The ashes are stored in the boiler room until the arrival of the ash truck. The elevator, which is of the bucket type, has a telescopic housing which is pushed up through the sidewalk opening and the chute extended to discharge into the truck. When the truck is filled and the elevator stopped, it is withdrawn below the sidewalk and the cover closed over the opening. The casing of the elevator is telescopic to allow of its retraction below the sidewalk, and the bucket chain is so designed as to allow the slack to pile up in an orderly manner at the bottom without becoming entangled.

19 Where conditions allow, an ash-storage hopper may be contrived within the building and a closed ash chute temporarily extended over the sidewalk to the ash truck. The storage hopper may receive ashes from a bucket car and whip hoist, a bucket elevator, or an air conveyor. It is not often that this method can be employed. The purpose of the building may entirely prevent

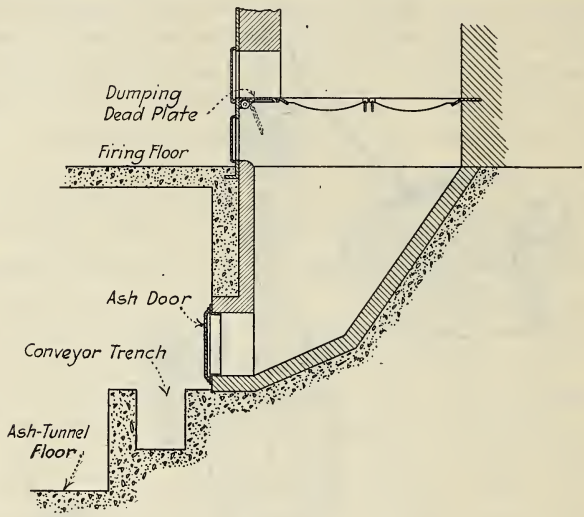


Fig. 7 Hopper Ashpit for Hand-Fired Furnace

the use of such a hopper, and the temporary extension of the ash chute over the sidewalk would not be permitted in the better or more crowded streets.

20 The first operation where labor can obviously be saved is in avoiding hoeing and shoveling ashes at the outset. This leads naturally to a consideration of the hopper ashpit.

HOPPER ASHPITS

21 The hopper ashpit is regular practice in the modern plant. Its size and general design will depend primarily upon the manner in which the ashes flow into it, the method of removal, and also upon the system of draft. The design begins at the top to suit the stoker.

22 In hand-fired, forced-draft anthracite plants the top of the hopper will conform to the whole grate area. With stationary grates fine ash is falling constantly over the whole area. As the fires are hand-cleaned, a dumping deadplate will keep the ash away from the firing floor and allow it to fall directly into the hopper, the forced draft being temporarily shut off. With dumping grates the ash will also fall from most of the grate surface. The ashpit will most probably be a storage hopper, which will be emptied

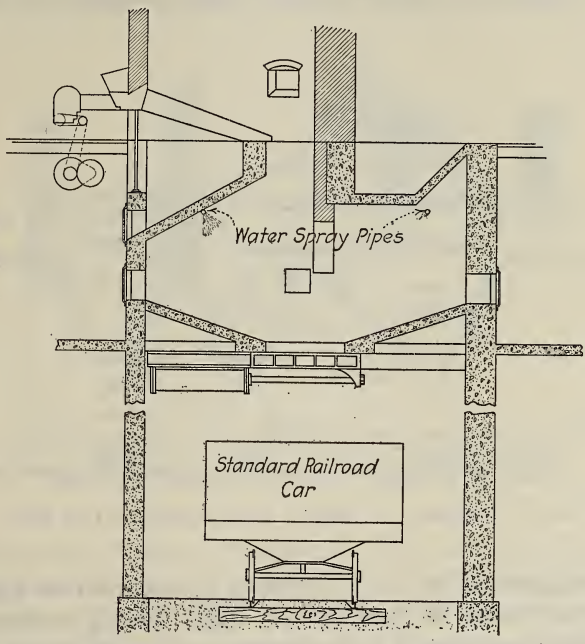


Fig. 8 Large-Capacity Hopper Ashpit

at intervals with the forced draft shut off. Large capacity is necessary because of the high percentage of ash in this fuel. The discharge door must be airtight to avoid increasing the cost of generating the forced draft and the possible reduction of ashpit pressure's lowering the boiler capacity. If it can be dumped directly into railroad cars, the basement being deep enough for this purpose, the hopper can be quickly dumped through bottom doors. Where conveyors are installed it is usual to use a side door and work the ashes out gradually into the conveyor to avoid choking it, as would happen with a straight dump. Fig. 7 illustrates a

layout of this kind. The conveyor will occupy the trench in front of the ash door. It may be mechanical, bucket or chain, or air or water, and is therefore shown schematically only.

23 With forward-travel underfeed stokers, chain-grate stokers, front-feed inclined stokers, and others the ashes are dumped at the rear of the furnace and the top of the hopper will be the width of the fire but of small dimension from front to rear to accord with the stoker dump. This reduces the capacity for storage as is seen by examining Fig. 9. Where large storage is imperative, the ashpit

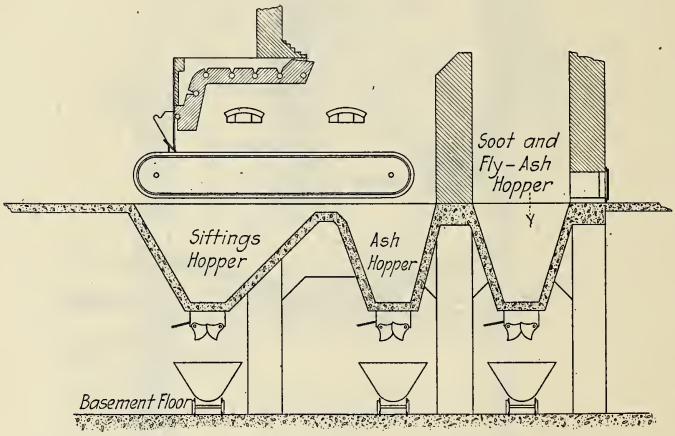


Fig. 9 Hoppers for Ashes, Coal Siftings, and Soot

may be designed as in Fig. 8, which is drawn from an illustration appearing in *Power* of January 17, 1922. Owing to the flatness of the bottom, some hand labor is necessary to effect complete discharge when desired.

24 With chain-grate stokers a hopper should also be provided for fine coal which sifts through the grates before ignition. Such an arrangement is shown in Fig. 9. This coal is dumped into some means of conveyance to be returned to the stoker coal hoppers.

25 Side-travel underfeed stokers dump at each side of the wind box for the whole depth of the fire. Hopper ashpits for this type of stoker may therefore have greater storage capacity since the top of the hopper extends to the front of the boiler. Fig. 10 illustrates an ashpit of this kind. In the Combustion Engineering Company's Type E stoker a valve is provided to admit

forced draft to the ashpit in order to burn out the ash more completely before dumping, and therefore the ashpit dump doors should be reasonably airtight.

ASHPIT CAPACITY

26 In discussing the amount of storage required, it will be noticed that there are frequently two places where the ash is stored: in the ashpit, and in the elevated ash bin if there is one. In some instances very little capacity may be sufficient, owing to storage

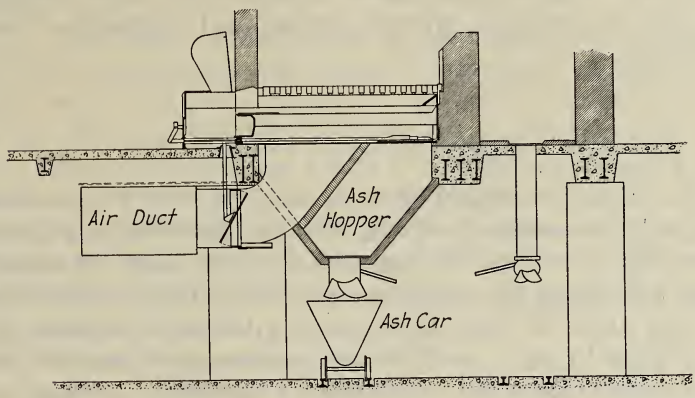


Fig. 10 Hopper Ashpit for Side-Feed Stoker

being taken care of outside the ashpit. The method of conveyance from the ashpit will have considerable bearing on the ashpit capacity necessary. But 24 hours' capacity should usually be provided in case of breakdown of conveyors, etc. Consequently some of the attributes of different methods of conveyance which bear on ashpit capacity should be mentioned here. The largest ashpit capacity will be requisite with conveying systems which are operated intermittently. These will include instances where ashpits are dumped into standard or industrial railroad cars, or into conveyors which might be choked by the direct dumping of an accumulation containing large clinkers and therefore need hand feeding or at least manual attention. Where a continuous-conveyor system is used the ashpit may be much smaller, and sometimes the ashpit becomes simply a funnel or chute to guide the ashes into the conveyor, as in Figs. 22 and 31.

27 Since the required capacity depends upon the amount of ash that accumulates between emptyings, it is controlled by the rate of firing and the percentage of ash in the coal. Multiplying the weight of coal burned between dumpings by the percentage of ash and allowing 40 lb. per cu. ft. gives the storage space required. Allowance must be made for unburned coal coming in with the ash, for neglect to empty regularly, for possible increase in the load and consequently in the rate of combustion, and for possible change to dirtier coal. The careful engineer will make the capacity of his ashpits perhaps 50 per cent greater than the calculations show, or even more. Also, sufficient space should be left above the top of the ashes in order to prevent burning and breakage of stoker mechanism.

HOPPER-ASHPIT DESIGN

28 The sides should slope at not less than 45 deg. in any case, and a minimum of 50 deg. with the horizontal is preferable. If one side is vertical the opposite side may have the minimum slope; but where two opposite sides slope, neither should slope less than about 55 deg. If the slope is too small, arching of the ash is likely to occur, and this will increase operating costs owing to the frequent necessity of encouraging the flow by breaking away the ash with bars poked up through the discharge doors. Where the width of the hopper would necessitate too great height to get these required slopes, the ashpit may easily be divided so as to have a number of discharge openings. An excellent example of reversed slope which results in absolutely reliable dumping is illustrated in Fig. 19.

29 If a very small slope is used so as to get large capacity such as in Fig. 8, access doors should be provided at the top of the slope so that the ashes may be pushed to the dump doors with a minimum of labor, and ample space should be left so that long ash tools can be wielded with ease.

30 Large discharge openings should be used, though their size depends to some extent upon the method of firing. When small anthracite is burned with sufficient steam in the combustion air, no clinkers are formed and the ash is in small pieces and easily dumped. Modern intensive firing of bituminous coal tends to cause the formation of large clinkers, and the openings should be large

enough to pass these without having to break them into smaller pieces to allow dumping. In earlier installations, door openings about 18 in. square and even less were common, while modern practice required 30 to 36 in. as a minimum, with many instances of clear openings 5 ft. square.

31 In some instances the discharge door is in one of the sides of the hopper as in Fig. 7, and the attendant hoes out the ashes either into cars or conveyor. Whenever it can be adopted, however, the bottom discharge is undoubtedly the least costly in labor and well repays the added expense of the greater height of basement needed.

32 The height of the bottom of the hopper above the basement floor depends upon the system of conveyance adopted. It is greatest where standard railroad cars are used, about 8 or 9 ft. to clear the cars and about 17 to 18 ft. if a locomotive must pass under. With industrial cars 5 or 6 ft. is sufficient, though a clear headroom of 6 ft. 6 in. to 7 ft. is preferable. With some conveyors it may be much less, but in any case there should always be sufficient vertical space to allow a bar to be pushed up into the hopper to clear away any obstruction. In designing new plants ample headroom should always be provided.

33 Where storage takes place in the ashpit, water spray pipes should be provided for quenching the ashes. These pipes should be near the top and sheltered from the incoming ash. Or a substantial spray ring such as illustrated in Fig. 14 may be used. Many prefer to make the hopper sufficiently large for the ashes to remain long enough to cool naturally. By allowing some ash to remain when dumping, for the new hot ash to fall on, the doors and lining are protected and the ash gradually cools without the use of water. If too much water is used it will leak from the dump doors and flow about the basement; and as it contains much fine ash in suspension, it will clog sewers and necessitate cleaning them frequently.

HOPPER-ASHPIT CONSTRUCTION

34 The hopper is usually worked into and suspended from the general structure at the firing-floor level, to avoid any supporting columns which would block up the basement thoroughfares.

35 Hopper shells have been made of sheet steel lined with

firebrick, but this construction is undesirable owing to rapid corrosion from sulphur in the ashes. Shells of reinforced concrete about 6 in. thick are common and satisfactory. The most modern construction, and probably the best method so far devised, is to use a structural-steel skeleton and make the shell of substantial cast-iron flanged plates bolted together. Corrosion is not troublesome because cast iron suffers much less than steel from this cause. This construction facilitates connection with the upper structure and with the dumping-door frames.

36 Owing to the heat of the ashes and the possibility of the combustion of unburned coal, the hopper shells should always be

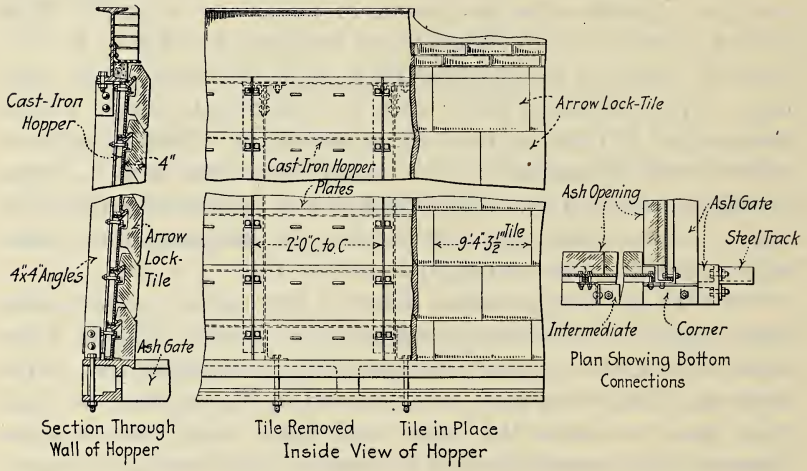


Fig. 11 Baker-Dunbar-Allen Ashpit

lined with firebrick, which may be of second quality. With proper quenching of the hot ashes with water sprays a lining of well-burned hard paving brick is very satisfactory.

37 The method of construction of the Baker-Dunbar-Allen hopper ashpit is illustrated in Fig. 11. A suspended skeleton of structural steel carries the hopper shell of heavy cast-iron flanged plates. The lining is of special firebrick blocks which will not spall under the temperature changes which occur. As is seen in the left-hand view, these blocks are hung from the shell and interlocked in a manner that prevents displacement but allows of easy renewal, and no mortar joints are used.

HOPPER-ASHPIT CLOSURES

38 Hopper ashpits will usually be provided with doors to retain the ash and prevent the passage of air, and to allow of the ash being dumped or hoed out at intervals as desired. With hand firing and natural draft the doors need not be airtight so far as draft is concerned; but with chain grates and with closed-ashpit forced draft the doors should be tight to avoid loss of air pressure and waste of blower power. Inleakage of air may reduce boiler efficiency in some cases by excess air passing the fire through the stoker ash dump; and when forced-draft

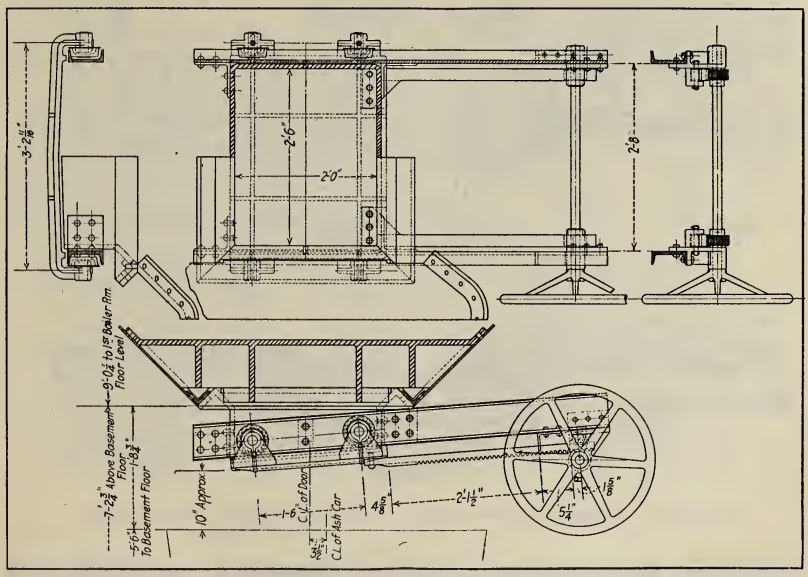


Fig. 12 Assembly of Ashpit Door and Frame

stokers are heavily loaded, gas may escape through the dumping doors. Where much combustible gets into the ashpit, either through carelessness or imperfections in stoker design, leakage of air through ill-fitting dump doors often causes sufficiently rapid combustion in the ashpit to result in damage to grates, hopper structure, or dumping doors. Airtight ash doors are therefore highly desirable in almost all cases.

39 There are several ways in which the dump openings may be sealed. The doors and faces may be machined, or the frame

may be provided with a groove packed with asbestos rope, while the door has a rib or tongue which is squeezed into the asbestos packing by a cross-bar and screw spanning the door. The latter method has been used with satisfaction in vertical doors like that of Fig. 7. The door and frame may sometimes be left unfinished with sufficiently well-fitting smooth-faced castings that will prevent much ingress of air. The doors should usually be lined with firebrick to prevent warping due to hot ashes lying on them soon after dumping.

40 An example of ash door is illustrated in Fig. 12. These doors are of substantial construction, and are carried on rollers

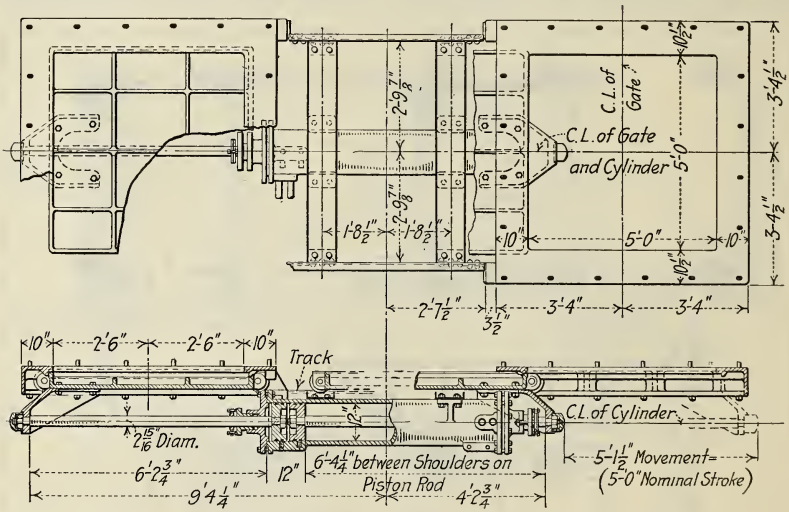


Fig. 13 Baker-Dunbar-Allen Power-Operated Ash Doors

running on steel members. They are operated by hand with rack and pinion. They were designed and installed in the station using the system illustrated in Figs. 17 and 18 in 1916, and continue to operate with entire satisfaction. Smaller doors are operated by hand, but when the openings approach 3 ft., power operation is advisable for speed. The larger hand-operated doors are worked by gearing such as rack and pinion, while power operation may be hydraulic, compressed air, or electric.

41 Fig. 13 illustrates two Baker-Dunbar-Allen dumping doors arranged for compressed-air or hydraulic operation. The cylinder is worked in between the openings and contains two pistons, one connected to each door. The doors are lined with firebrick.

42 The Diescher dumping door is shown in Fig. 14. This door is provided with rollers which run on a curved track. Owing to the curvature of the door in conjunction with the flanges which carry the rollers, it retains water to form a seal, and this water renders lining unnecessary. Of course, when ashes are dumped the seal is lost; but as more water enters through the sprinkling pipe the seal is quickly remade. The door can be run off the track and another door run on if replacement is ever necessary by simply removing

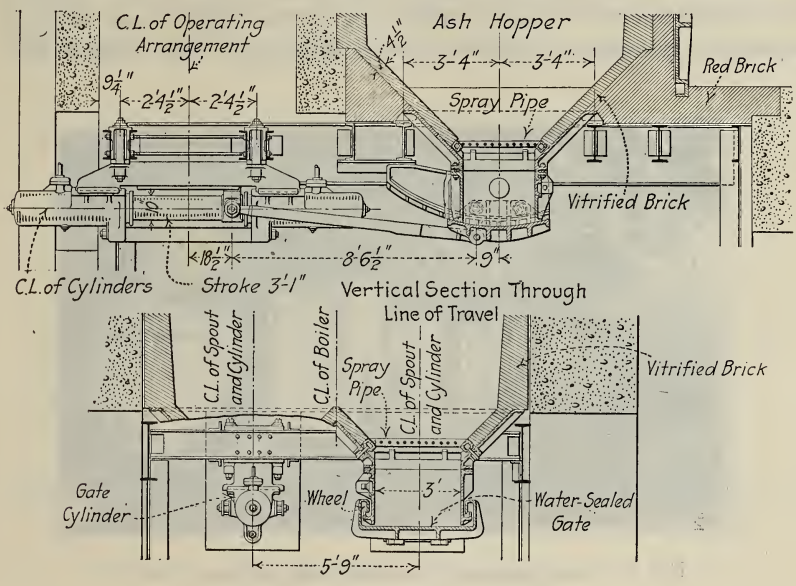


FIG. 14 DIESCHER POWER-OPERATED ASH DOORS

the pin of the driving connection. There is ample power and strength to shear easily through any clinkers which may get caught during closing.

43 Fig. 15 is a view of the basement of the boiler room of the Kelly-Springfield Tire Company, where the hopper ashpits consist of heavy cast-iron flanged plates bolted together and to the frames of the Diescher doors. The operating valves are above the concrete pads at the right. These pads are provided to protect the operating valves from ash cars being maneuvered near them. In this instance the basement floor has been temporarily raised owing to the use

of small ash cars which dump the ashes for filling about the premises. Whenever it becomes necessary, the basement floor level will be dropped to accommodate standard-gage railroad cars.

44 Piston-operated doors may use steam or compressed air, or may be operated from the feed line at some point where it will not interfere with correct recording of water fed to boilers, or from the city water supply; or a pump may be installed especially for this purpose to supply water at about 100 lb. pressure. If boiler feedwater is used, care must be taken to use hot-water packing in cylinders and valves. Instead of piston drive they may be operated by electric motor through gearing. The rugged con-

Fig. 15 Ash Basement of Kelly-Springfield Tire Co.

struction of these power-operated doors to meet their severe service, together with their size and elaboration, shows very striking differences from comparatively recent practice in large power stations, and clearly indicates the careful thought which has been given to this subject in recent years.

45 Instead of using doors, closure may be effected by water seal. This method may be described as a U-tube with one leg forming the hopper ashpit while the other is open to the atmosphere and possibly to the entrance of a clamshell bucket. Water in the bend of the U-tube forms a seal and serves to quench the ashes.

46 An excellent example of this method is illustrated in

Fig. 16, which is a cross-section of the boiler room of the Springdale Station of the West Penn Power Company. The operation of this system is very simple indeed. More than sufficient water for the seal is provided by the waste cooling water from the clinker grinders. The overflow is from one ashpit to the next, until it is finally discharged from the last ashpit.

47 This system of ashpit requires considerable depth of basement: not less than about 30 ft. from firing floor to basement floor. At Springdale it is 35 ft., and as deep excavation for foundations is necessary, extensions will probably have much deeper ashpits. These deep ashpits provide excellent storage. Actual test shows

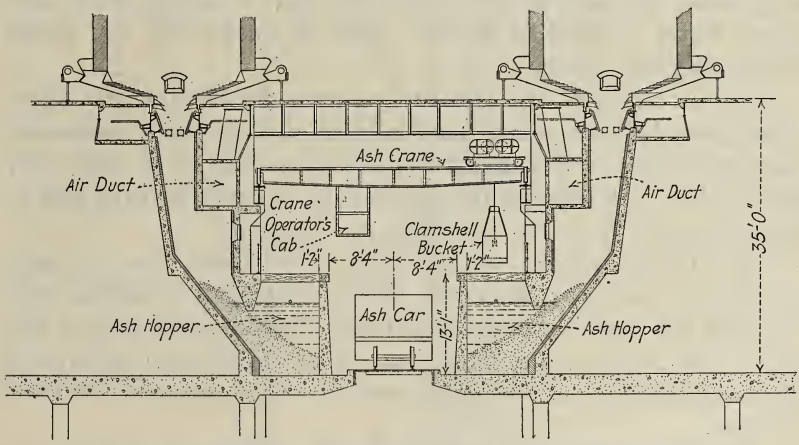


FIG. 16 WATER-SEALED ASHPIT

the present ashpits to have a capacity of three days' operation at 200 per cent continuous rating of the boilers.

48 Removal of ashes from the ashpits is by traveling crane and clamshell bucket, and is one of the most comfortable and easy jobs around the plant. One crane operator handles the ash output of the station in two hours per day.

49 A further advantage of this method is that no combustible or corrosive gas escapes into the basement. When boilers are being pushed with heavy loads, gas often escapes from hopper dump doors. Apart from discomfort and danger to those in the basement, considerable corrosion of ironwork often occurs. Such troubles are entirely obviated with the water-seal ashpit.

MECHANICAL CONVEYANCE

50 The method of conveying ash by emptying ashpits into small dumping cars has a great deal to recommend it. The cars are inexpensive, so that one or two idle spare cars are not objectionable, and the cars can then be repaired without interrupting ash conveyance in the slightest degree. The cars can be moved by men, animals, tractors, or locomotives; and they can be run about the floor or on tracks, all according to the amount of ash to be moved and other conditions. They can be run about the premises and dumped to fill depressions, or lifted on elevators to be dumped into an elevated ash bunker, or be dumped into the bucket of a whip hoist. In installations of this kind there is always much dust, unless water is supplied in the hopper to quench the ash before discharging into the cars.

51 In a subject of this nature wherein classification and direct comparison between plants and methods are virtually impossible, instances where advantageous changes have been made have very great value. The authors feel particularly favored in being able to draw attention to one or two cases of this kind.

52 In the Ashley Street Station of the Union Electric Light and Power Company, St. Louis, Mo., a system of industrial railway and ash cars was installed in 1905. The basement floor of this plant is 30 ft. below the flood stage of the Mississippi River, and this necessitated a watertight basement with consequent elevation of ashes.

53 The boilers are equipped with chain-grate stokers discharging ash into hoppers having a capacity of 24 hours. These hoppers are of steel plate lined with vitrified brick laid in cement, and each has two large horizontal dumping doors which are illustrated in Fig. 12.

54 The railway is 30 in. gage and the dumping cars have a capacity of three tons each. The cars were drawn by mules, and two men were required to load the cars and drive the mules. The cars were run on to either of two Otis elevators having a vertical travel of 120 ft., and two other men were required to run the full cars on to the elevator and remove the empties. Two more men at the top ran the cars off the elevator and dumped them into the ash bunkers. In all, six men were required for each of the two day shifts and four men during the midnight shift, making a total of 16 men

to handle the ash. The large number of men required to operate the system was mainly due to the fact that the men were not allowed to travel on the elevators, which were inspected for freight only. The elevated ash bunker had a capacity of eight railroad cars, which was about two days' production. The total storage capacity was therefore sufficient for three days' production of ashes,

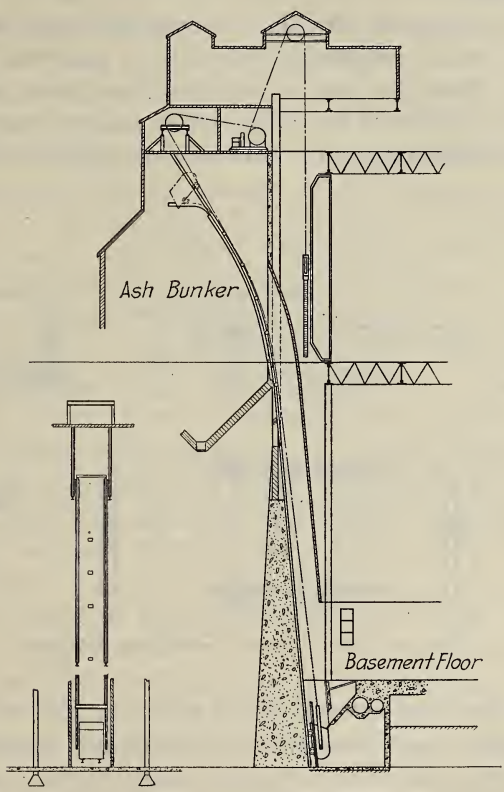


Fig. 17 Skip Hoist

one day in the hopper ashpits and two days in the elevated ash bunker. To reduce the operation and maintenance costs, the use of the elevators and of mules was discontinued in 1916. The elevators were replaced by a bucket worked by a skip hoist and a storage-battery locomotive used to haul the ash cars.

55 This was one of the first installations of a skip hoist to handle ashes. The mechanism of the hoist was supplied by the

Otis Elevator Company, and the general arrangement is illustrated in Fig. 17. The same industrial railway was used with some small rearrangement of the track to bring the cars to the hoist. An excavation was made in the basement floor to accommodate the bucket of the hoist and the dumping hopper, and this required careful thought and work to avoid any disturbance of the circulating-water ducts.

56 The bucket is carried on four single-flanged wheels which run on two double-rail tracks of 5 ft. 3 in. and 6 ft. $1\frac{1}{4}$ in. gage, respectively. There is a pair of wheels at the bottom of the bucket and another pair at the top. The lower pair run on the broader-gage track and the upper pair on the narrower gage. At the top end

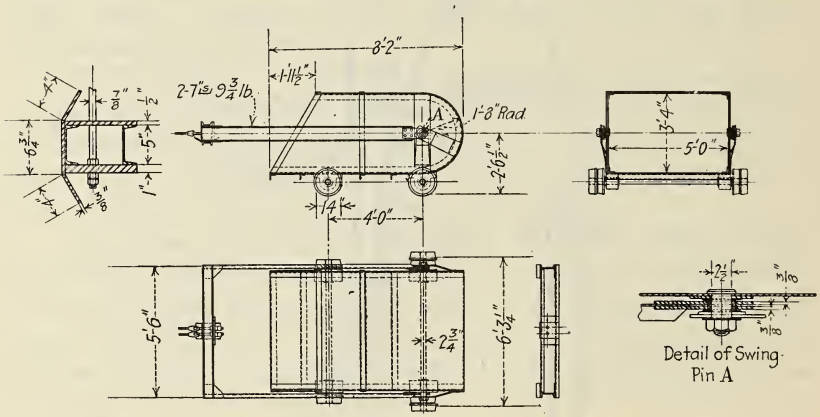


Fig. 18 Details of Skip-Hoist Bucket

of the run the narrower-gage-track rails are bent over as shown, so that the upper part follows the bent track and ceases rising, while the lower part follows the straight, nearly vertical, track until it becomes the higher. Thus the bucket turns over and empties into the hopper. It is counterbalanced by a weight having half the travel of the bucket, by carrying the balance rope around a sheave attached to the weight, as may be seen in Fig. 17. The details of the bucket are presented in Fig. 18.

57 In changing over from the elevators to the skip hoist, it was necessary to build a new elevated ash bunker to work in with the alterations in the new general layout.

58 The operation of the skip hoist is mainly automatic.

When an ash car has been dumped into the bucket, a switch button is pressed. This starts the hoisting motor, which raises the bucket to the top of its travel where it turns over and discharges into the elevated bunker. A switch trip at this point reverses the motor and thus lowers the bucket to its starting point, where another switch trip stops the motor. One pressure of the button therefore results in hoisting and dumping the bucket and in returning it to the starting point ready for another load. A spring safety bumper is placed 6 in. below the regular bottom stopping point.

59 The operation of this new system is entirely satisfactory and it has given good service. It has reduced the labor cost of handling ash one-third and the cost of maintenance one-half.

60 Experience with the storage-battery locomotive shows that in the heat of the basement the electrolyte evaporates rather rapidly, resulting in some battery trouble. A gasoline locomotive is now used and gives entire satisfaction.

61 The corrosive vapors in ash basements which come from ill-fitting dump doors when forced-draft boilers are heavily driven, and from exposed partly quenched ashes, are very destructive of steelwork. In many cases it has been found necessary to coat structural steelwork with gunite, and in modern stations concrete instead of steel in basements is usual. The health and comfort of the workers therein should also be considered, if only from the standpoint of economy, because labor will be easier to get and will be more efficient.

62 In the original system installed at the New Bedford Gas and Electric Light Company's plant, the ashes collected in hopper ashpits were emptied into Koppel dump-body ash cars. These cars had a capacity of 36 cu. ft. each, and were propelled by man power along a cast-iron track set in the concrete floor. The cars were pushed out into the yard and dumped into a large pit. When this pit was filled, a locomotive crane dug out the ashes with a clamshell bucket and deposited them in an elevated concrete bunker. This bunker was so arranged that trucks could drive under and receive the ashes through the dumping door. The ashes were given away to the city and otherwise, and no expense was involved in their disposal after they had been put in the elevated bunker.

63 The use of the manually operated ash cars has been recently discontinued. A dump body of 40 cu. ft. capacity was adapted to a storage-battery truck, and the operation of the elec-

tric truck immediately eliminated four or five ash laborers. The large pit with the locomotive crane and grab bucket was also discontinued. An automatically controlled skip hoist with the skip car normally in a small concrete pit just large enough to hold the car, was installed about 30 ft. outside the boiler room. The truckman pushes a button switch which starts the elevator motor, and this raises the car, dumps it into the ash bunker, and returns it to the concrete pit ready for another load. The hoist, car, and

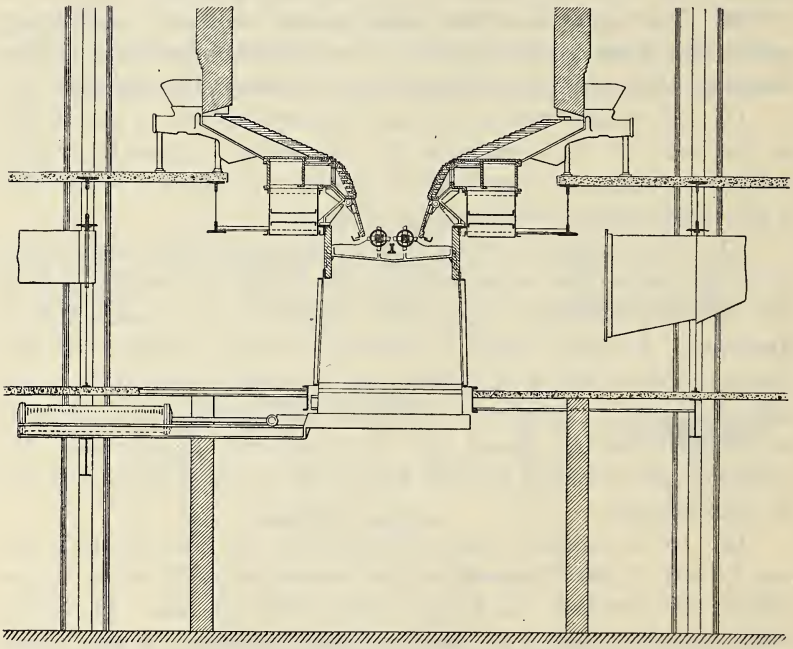


Fig. 19 Hopper Ashpit for Dumping into Standard Gondola Railroad Cars

electric controls were furnished by the Geo. T. McLauthlin Company, of Boston, and have given entirely satisfactory service. Two men now handle the ashes from the station, which has a load of about 38,000 kw. at the present time. The cost of maintenance of the electric truck has been comparatively low.

64 Fig. 19 illustrates a hopper ashpit arranged for dumping directly into standard gondola railroad cars which form part of the equipment of the station. The hoppers are lined with brick, not shown, and have a capacity of about 2500 lb. of ashes. The stokers

are equipped with clinker grinders. The ash gates are of the sliding type and are operated by compressed-air cylinders. Water spray pipes are provided near the top of the hoppers for wetting down the ashes before dumping.

65 The present consumption of this station is 413,000 tons of coal per year. The ash content of the coal by analysis is 11 per cent, and the produced ash contains 13 per cent of combustible. The yearly ashes from the station amount to 52,800 tons.

66 The ash cars are handled by an electric locomotive and are dumped into an outside pit, from which they are recovered by a crane and grab bucket and loaded into the purchaser's trucks. The electric locomotive and the crane are also used for coal handling, and careful time study shows that 25 per cent of their time is devoted to ash handling. Ashes were dumped once during each shift, so that three shifts of ash handlers were necessary. These ashes were all sold and the proceeds very nearly paid the labor cost of handling them.

67 The estimated cost of the necessarily increased height of the building due to this method of ash handling is naturally subject to comparison with some other system and is therefore not of value for general comparison.

68 It is estimated that 45 lb. weight of compressed air is required per day per boiler to operate the hopper-ashpit dumping doors. From recent tests it is found that 1030 gal. of water is used in spraying one ton of ashes, and this amounts to about 4 per cent of the total general-service water used by the station.

69 Bucket conveyors such as the Peck carrier have been extensively used. With the chain and buckets forming a ring system, the conveyor is often used for coal in the daytime and ash at night. The coal is received from crushers if necessary at one end of the building and elevated to a point above the bunkers into which the buckets are discharged. The ashes are dumped into the buckets during their lower run under the ashpits, elevated, and dumped during the upper run into an ash bin which can be discharged into railroad cars or motor trucks.

70 Fig. 20 illustrates a Peck carrier installation by the Link-Belt Company in a downtown heating station where coal and ash have to be handled by motor truck. The building occupies the entire site, and no projections of any kind are possible. The basement floor is 12 ft. below the alley level, and as the alley is only

15 ft. wide it was necessary to build a recess where the trucks could drive in and dump coal, and then load up with ashes.

71 The conveyor is located centrally between the boilers and handles coal as well as ashes, and also the siftings from the chain-grate stokers. The run-of-mine coal is screened as it is dumped from the truck, the screenings going directly through the feeder to the conveyor, and the lumps passing through the crusher.

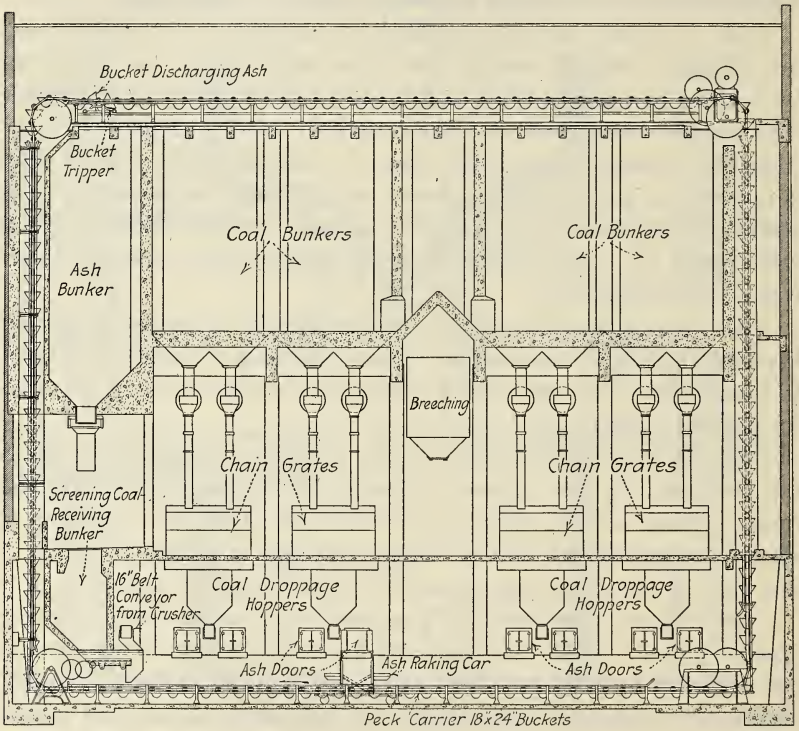


Fig. 20 Peck Carrier Ring System

The vertical lift is 66 ft. and the horizontal centers are 85 ft. The overlapping buckets are 18 in. wide and 24 in. on centers. A 7.5-hp. motor drives the conveyor at a speed of 45 ft. per min. The buckets are discharged into the overhead bunkers by a regular tripper on the upper run.

72 The coal bunker is continuous along the boilers. The ash bunker, which has a capacity of 130 tons, is located in line with and immediately adjacent to the coal bunker and at the same elevation.

Consequently the buckets discharge coal into the coal bunker or ash into the ash bunker according to the position in which the tripper is placed. Also, the ash-bunker discharge is immediately above the recess into which the trucks are driven, just as the coal-receiving hopper is immediately below it. Therefore the trucks, which have automatic dump bodies, drive into the recess over the

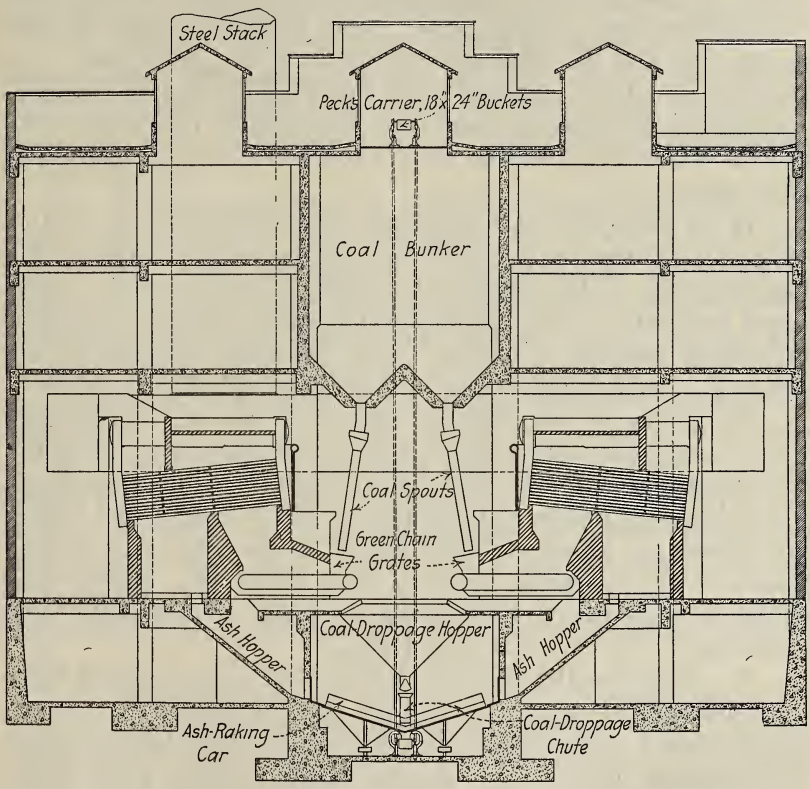


Fig. 21 Cross-Section of Boiler Room with Peck Carrier

coal-receiving hopper, discharge their load, and receive ashes without changing their position. The ash gate is operated from the floor level. All coal and ash hauled to and from the plant must necessarily be carried in large trucks. Quick unloading and reloading therefore materially increases the road time of the trucks; and this is a considerable item in the downtown district where traffic is more or less restricted.

73 A novel method is employed for the removal of the ashes from the ash hoppers to the conveyors. A double raking apron is contrived as a car mounted on tracks and straddling the conveyor. It is movable the entire length of the boiler house and can be brought in front of each ash door, forming a continuous chute from the hopper door to the conveyor. The operator stands on one side of the car and pulls the ashes down into the chute. After a flow of ashes has been created, water under pressure is used to keep up the flow, and the ashes are removed in a very short time. The cross-section of the plant shown in Fig. 21 illustrates this clearly. Considering the location and the surrounding restrictions, the general arrangement proved ideal for the purpose and has given good service.

74 A great deal of trouble can occur with conveyors of this type when allowed to reverse with a loaded chain. Although the manufacturers of the equipment provide a safety device, it was found necessary in this installation to devise a positive means of preventing reversal of motion. This has been accomplished by means of a band brake similar to those used on hoisting engines.

75 Owing to the abrasive nature of ash the maintenance cost of mechanical conveyors is high. The ashes grind away the connecting pins, and even with regular renewals the pins sometimes wear excessively and cause breakdowns. With a ring bucket system breakdowns are sometimes quite serious, as the ashes may have to remain in the basement for some days before the conveyor can be got to work again; and keeping the plant running under such conditions is very far from comfortable.

76 When handling coal and ashes in the same conveyor there is the difficulty of the fine coal and wet ashes packing in the buckets, so that there is a needless continuous load on the equipment with a corresponding reduction in carrying capacity. The water which leaches from the ashes is exceedingly corrosive and helps to carry the fine abrasive ash into the wearing parts.

77 The life of these conveyors is considered to be about seven or eight years, and extensive repairs must be made every two or three years. The excessive cost of maintenance of ash conveyors of this kind has led to their replacement in some installations with electrically operated cars carrying the ashes to outside pits from which they are loaded into railroad cars with a bridge crane, and the cost of handling has been reduced by 50 per cent.

78 Hopper ashpits with large doors should not be dumped directly on to chain or bucket conveyors. To avoid choking or swamping them, an apron or auxiliary hopper may be used with advantage. With ashpits such as that shown in Figs. 4 and 21, there is no such danger as the ashes are delivered more gradually. With direct dumping, large clinkers are liable to jam and cause breakdown, and opportunity should be given for breaking them.

79 The Fisk Street and Quarry Street stations of the Commonwealth Edison Company of Chicago are each equipped with bucket conveyors which handle both coal and ashes. There is an additional pair of ash conveyors at Quarry Street which carry the ashes to the train shed.

80 The apron conveyor is a modification of the bucket conveyor and consists of flat overlapping sheets with side flanges attached to a chain and forming a continuous steel-belt conveyor. Four such conveyors were intended for installation at Fisk Street, Chicago. Three were installed and put in service, but the cost of operation and maintenance proved so high that the fourth was never installed. The parts of the fourth were used to repair the other three. An electrically operated ash car is now used in place of the fourth, and the other three apron conveyors are being replaced with similar ash cars. The cars are dumped into a pit outside the building. A bridge crane takes the ashes from the pit with a grab bucket and loads them into railroad cars. The cost of maintenance as compared with the apron conveyors is almost negligible. The outside pit and crane were originally installed for and used with the apron conveyor.

81 At Northwest and Calumet stations, Chicago, the ash is dumped from hopper ashpits directly into railroad cars. The only maintenance required is upkeep of ashpit linings and dumping doors. The cost of the ash-handling system at these stations cannot well be estimated because there is none. But the buildings must necessarily be higher to get head room under the ashpits, and the increased cost of the greater height is obviously chargeable to ash handling.

82 An example of a simple chain and cross-bar conveyor is illustrated in Figs. 4 and 5. The conveyor installed at one of the municipal plants of the Poplar Borough Council in England[1] and illustrated in Fig. 22, may be considered as a development of

[1] Described in *The Engineer* (London), August 5, 1921.

this idea in combination with the water seal of Fig. 16, though there is no ashpit storage. In fact, there is no ashpit, only sheet-steel unlined chutes ending in cast-iron nozzles which dip below the water level in the conveyor trough, thus providing the water seal.

83 The conveyor is in duplicate so that one can always be in use in the event of breakdown of the other. The conveyor trough, which is of cast-iron plates, is divided by other cast-iron plates along the center. Each channel houses two chains connected by cross-bars which drag the ashes along the bottom. The return of the chain is also below the surface of the water. The transverse centers of the chain are 19½ in. apart and the cross-bars

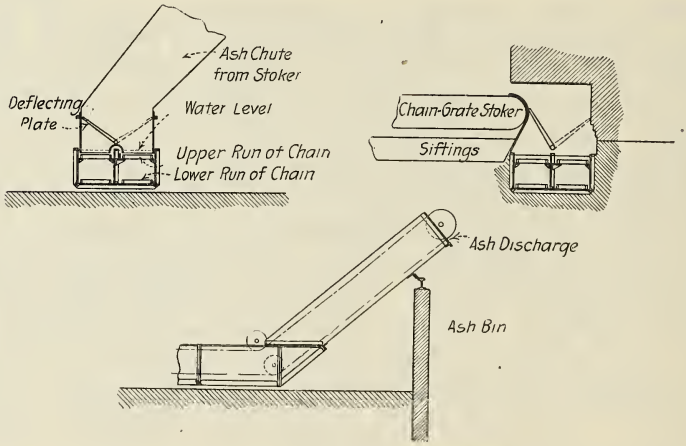


Fig. 22 Water-Seal Chain and Cross-Bar Conveyor at Poplar, England

are 25½ in. pitch. Therefore large clinkers easily fall between the upper run of the chains and cross-bars to the bottom of the trough. The end of the trough is carried upward at an angle of 41 deg. to its discharge into the ash bin, from which the ash is recovered by grab bucket.

84 The views of this conveyor shown in Fig. 22 will serve to give a good idea of its arrangement. The upper left-hand view shows a general cross-section of the trough. The water level is maintained close to the top by ball valves on the water-supply line. The double cast-iron nozzle on the end of the ash chute dips below the water to form the seal, and is provided with a deflecting plate

which may be swung over so as to guide the ashes into either conveyor as desired.

85 The upper right-hand view is of particular interest in that no vertical space was available, not even for an ash chute. This chain-grate stoker was provided with a guard plate around the rear end, and the deflecting plate may either be against this guard plate and cause the ash to be discharged into the right-hand trough, or against the bridge wall so as to discharge into the left-hand trough.

86 The lower view shows the discharge end of the conveyor. The channels are continued above the water surface by an open-top double trough of sheet steel which carries the driving sprockets at its upper end. Idler sprockets carry the chain around the angle. The chain-tightening gear is at the other end of the trough.

87 The chain is made up of drop-forged links, alternate links having integral hubs which enter the other links and so provide large wearing surfaces. After six months' use no sign of wear or corrosion was apparent. It is of interest to note that corrosion is not expected because the water in the trough becomes alkaline rather than acid. The quantity of make-up water required is about 40 per cent of the weight of the ashes handled.

88 No ash crushers are necessary as the space between chains and cross-bars is sufficient to allow large clinkers to fall through the upper run of the chain into the water; and the hot clinkers break up on falling into the water. It is reported that the ashes as delivered from the conveyor are in a finely divided state, the average size of the pieces being about equal to that of a pea. They thus form an excellent aggregate for cinder concrete.

89 The length of this conveyor is about 140 ft. and it is designed for a chain speed of 20 ft. per min. One conveyor running at 9.5 ft. per min. and handling 3.25 tons of ashes per hour takes a little under 2 hp.

90 In some other installations only a single conveyor is used with auxiliary chutes for use in case of breakdown. They are constructed by the Underfeed Stoker Company, Limited, of London, England, who own the British patents for the system; and by the Combustion Engineering Corporation, who own the American rights.

91 In addition to the ashes from the main ashpit, the fly ash and soot from the boiler setting must also be cared for. Sometimes a chute to the main ashpit will meet the case, but care must be

taken to close the chute with a suitable valve or damper to prevent air passing into the boiler setting. Reference may be made to Figs. 9, 10, 21, 24, and 26 for methods of dealing with fly ash; and the chute for this purpose may be seen between the operating cylinders of the dump doors in Fig. 15. It is one of the outstanding advantages of the air conveyor that it can be piped so easily to handle soot hoppers in boiler and economizer settings, flues, chimney base, etc.

FLUID CONVEYORS

92 The salient advantage of fluid conveyors is that there is no mechanism to wear out, which is very important owing to the abrasive nature of ash. This does not mean that there will be no wear, but as the worn parts are stationary, an enormous amount of wear may take place before renewal is necessary, and such renewal is generally very easily effected.

93 As only two fluids are economically available, such conveyors may be classified into air conveyors and water conveyors. In either case, the fluid must have sufficient velocity to propel the ash; and since this velocity depends upon the density of the fluid, it is obvious that the velocity in air conveyors must be very much greater than in water conveyors. As the velocity is so high in air conveyors, the abrasive effect is also high and the wear heavy. Especially is this the case at elbows, and inexpensive wearing backs must be provided which can be easily renewed. It is to be expected that more power will be expended to move the ash at the high velocity necessary with the air conveyor, since the momentum given to the ash is so much lost energy.

AIR CONVEYORS

94 Air is passed through a pipe at a sufficiently high velocity to carry the ashes along with it. The air is admitted at one end and ash intakes are provided wherever required. There are two systems of generating the air current. In one, the pipe outlet is connected to an ash-storage tank in which a vacuum is caused by means of a steam-jet or mechanical exhauster. In the other, the air current is induced by a steam jet between the ash intakes and the outlet.

95 A typical layout of an air conveyor with vacuum storage

tank is illustrated in Fig. 23. A steam-jet exhauster is attached to the top of the tank and may discharge into the atmosphere or into the chimney or a silencer to reduce the noise. A mechanical exhauster may be used instead of the steam jet because only a little of the ash dust passes through it, and this may be reduced to a minimum by the judicious use of water sprays. It is probable that this system takes a little more power to operate, because the average density of the air in the conveyor pipe is low, and a greater weight of air is therefore required to convey the ash. But this may easily be so overshadowed by other advantages as to be quite negligible.

96 As the highest velocity which can be attained by particles

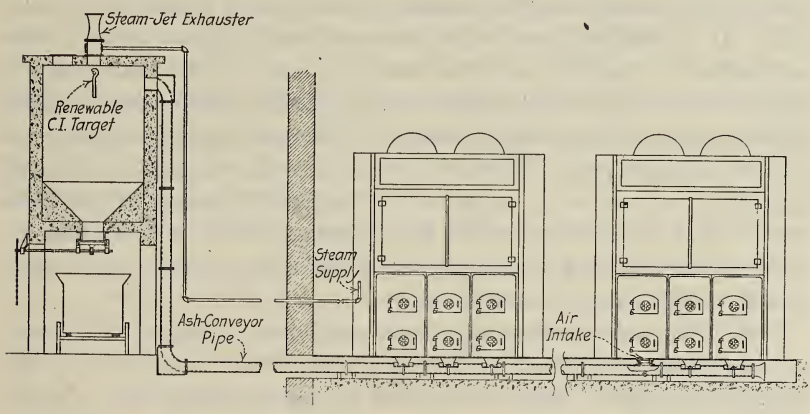


Fig. 23 Air Conveyor with Vacuum Storage Tank

of ash is approximately that of the air, the wear seems likely to be less than where there are steam jets in the conveyor pipe. It is claimed that ordinary gray iron piping may be used, and a test is offered wherein a piece of conveyor pipe was replaced with light wrought-iron pipe which showed but little wear after a year's service carrying 5000 tons of ash.

97 It is obvious that the steam jet, which is located in the conveyor pipe of the other system of air conveyor, could not be replaced by any kind of mechanical blower because the passage of the ash through it would quickly destroy it. An ash tank is not required as part of the system. The conveyor pipe may discharge into the atmosphere and be arranged to fill up depressions in ad-

jacent ground. But unless the ash is sprinkled with water before it leaves the conveyor pipe, the resulting dust may be objectionable. The conveyor pipe may discharge directly into railroad cars, though this is usually inadvisable because of the difficulty of not having an assured supply of empty cars and of the ash car blocking the track while being filled and possibly interfering with the use of the track for coal cars. It is usual for the conveyor to discharge into an elevated ash tank supported on columns, so that carts, motor trucks, or railroad cars may be run underneath the tank and be filled quickly by opening the ash valve in the bottom. If a storage hopper is used, a vent must be provided for the escape of the air.

98 The ash particles may momentarily attain velocities approximating to that of the steam jet at a little distance from the muzzle of the nozzle and therefore local abrasion may be considerable. It is usual to locate the motor jet at an elbow, as it is then convenient to aim the jet in the new direction. When the conveyor pipe is very long, extra nozzles are installed in some cases. These extra nozzles are sometimes located in elbows and sometimes in a run of straight pipe. When the nozzles are arranged in straight pipe they are set at an angle to the axis of the pipe and are then not so efficient as they would be if coaxial. It is not possible to place nozzles coaxially because they would be in the path of the ash, which would quickly destroy them.

99 The ash openings may be arranged in any position desired, such as in the firing floor in front of hand-cleaned ashpits, near the bottom of hopper ashpits, connected to pipes to draw soot and fly ash from the later boiler passes, etc. They will serve several lines of boilers as easily as one, and such lines do not need to be parallel but may be at various angles and at different levels. An air inlet is provided at the beginning of the pipe, for it must be remembered that it is the velocity of a large body of air which carries the ash, and not the vacuum. The inlet openings that are not in use should always be kept tightly closed with the plugs provided.

100 The speed with which ash is carried away by air conveyors of either type is surprising to any one who has not seen them in operation. It is not possible to choke an intake by as many men as can work around it shoveling and hoeing ash into it as quickly as possible, without heaping over the opening. Hence economical ash handling by air conveyors is largely dependent upon the convenient location of the intakes and the facilities for the most

rapid feeding possible without choking the intakes with clinkers. They will usually carry anything that will pass through the intake opening.

101 As an instance of the speed at which ashes can be handled, a plant may be cited containing fifteen 500-hp. boilers equipped with Murphy stokers, where the average amount of ashes in each pit ranges from 1000 to 1300 lb. One man cleaned out eight ashpits in 27 min. from the time steam was turned on until it was turned off.

102 They should not be directly connected to hopper ashpits, however; such ashpits should discharge into auxiliary hoppers or funnels with stout gratings to hold back clinkers of such size as should be broken up before reaching the conveyor intakes. With modern intensive firing much higher furnace temperatures are carried than formerly, and this results in the formation of clinkers of large size. Some stokers are provided with clinker crushers, and when this is not the case it is becoming increasingly necessary to add clinker crushers as part of the ash-handling equipment, so that the clinkers may be reduced to such size as the conveyors can handle without choking or breakdown.

103 The maximum capacity of a 6-in. conveyor is about four tons of ash per hour; that of the 8-in., six to nine tons; and of a 9-in. conveyor, ten to fifteen tons and even twenty tons in some cases. The capacity depends largely upon the size of the pieces. Ash should not be wet or quenched when fed to an air conveyor.

104 Ashes can be conveyed by air conveyors through a horizontal distance of about 500 ft. and through a rise of about 100 ft. It is often better to use several conveyors, either with a common ash bunker or a separate ash bunker for each, rather than a single conveyor with several branches. The capacity will be greater and also the efficiency, because there will be less leakage from idle intakes and through isolating dampers. The conveyor pipe may be run in any direction, up or down, and around any number of corners. In this respect it possesses a flexibility of installation which is not approached by any other system, and it can be adapted to circumstances which no other method can meet.

105 The cast iron used for pipes and elbows and other fittings is generally made of the hardest possible white iron, such as is not machinable and can only be ground, so that connections are commonly made by means of bolts slid into open lugs. The wear is

greatest at elbows where the ash direction is changed, and it is usual to provide "wearing backs" of easily replaceable blocks.

106 As the ash is projected from the pipe at high velocity, some kind of easily replaceable target is provided in or near the ash hopper. The ash strikes this target and falls without horizontal velocity, for if it were directed at the opposite side of the tank, it would soon cut a hole through it.

107 Air conveyors use a large quantity of steam while running; but as they remove the ash very rapidly, the cost of steam per ton of ash removed is quite small when they are properly operated. If the conveyance of ash is interrupted while the steam is blowing, the waste of steam will naturally be very great and the cost of steam per ton of ash will rise very rapidly. If the cost of steam per ton of ash with a reasonable method of operation is recorded, then if the ash is only fed half as fast, the cost per ton will be doubled, because the nozzle discharges the same quantity of steam in either case. As this is a matter of seconds, it may easily become of great importance. Suppose the ash handler turns on the steam and then walks some distance to get a hoe or shovel and then to the point where he is going to work on the ash. Then more time may be lost in preparing to work at another ashpit. When all the ash has been blown away, there is the walk back to the valve to shut off steam. The steam may also be blowing while some interesting conversation is taking place instead of hoeing ashes. All this will result in enormous waste of steam. Therefore methods of operating the steam valve from the vicinity of the ash intakes are very valuable in saving steam.

108 The Green Engineering Company uses an electrically controlled steam valve. A foot-operated switch is placed in a convenient position near each ash intake, and the switch is connected to a balanced steam valve which is operated by a solenoid. Steam is then blowing only while the attendant is actually at work at any particular intake.

109 The United Conveyor Corporation's main steam valve is operated by a triple valve. A small valve which is opened and closed by a quarter-turn is located near each intake. When all the ash has been hoed into the conveyor, the small valve is closed and steam consumption ceases. When the operator has swept up the floor into the intake, the small valve is opened momentarily to remove the sweepings.

110 Considerable expense ensues if steam nozzles are allowed to wear excessively before renewal. Not only is the steam consumption greatly augumented, but the increased energy of the larger steam jet results in higher air and ash velocities with greater wear of pipes, elbows, targets, etc. Such wear of steam nozzles is greatest with wet steam. These conveyors are only used intermittently and as the steam nozzles are often so located as to require a long line of supply pipe, they are usually supplied with wet steam.

111 Several methods have been developed to guard against the continued use of excessively worn nozzles. One is to provide a local reduction of wall thickness near or at the throat so that comparatively slight wear will cause the thin wall to break through. The resulting steam leakage is arranged to blow a whistle and so warn those interested that a new nozzle should be put in. Another method is to make the entire nozzle with thin walls, about $\frac{1}{32}$ in. When much wear takes place the nozzle bursts and puts the conveyor out of commission, and warns the attendant at the same time. A new nozzle is inserted in a few minutes.

112 In some instances the steam-jet conveyor is objectionable because the steam used represents so much water lost, and which must be replaced by "make-up." In modern central power stations boilers are loaded to 300 or 400 per cent of their rating during peaks, and the greatest care is necessary to avoid impure water entering the boilers under these conditions. It is becoming increasingly general practice to distill all make-up water so as to prevent any scale-forming or foam-making salts from getting into the boilers. Condenser leakage is diligently looked for and eliminated. Therefore, owing to the care with which pure water must be conserved, it is obvious that any apparatus which removes water from the system, such as steam jets whose steam does not eventually reach the condenser, will usually be frowned upon by operating engineers.

113 Of course, the matter entirely depends upon what proportion the steam so used would bear to the regular amount of make-up used by the station. One ton of steam will move four to eight tons of ash. The steam consumption depends upon the length of the conveyor, the height through which the ash is lifted, the number of bends, and the care used to economize steam during operation. With a coal containing 12 per cent of ash, two tons of steam would be used per 100 tons of coal. Taking an average evaporation of

9 lb. of water per pound of coal, the conveyor would use two tons of steam out of each 900 tons generated or nearly 0.2 per cent. To allow for careless operation or negligent maintenance and other contingencies, it would perhaps be advisable to allow say 0.3 or 0.4 per cent in arranging for extra distillation for make-up. If the make-up for the station is 2 per cent, then the distilling capacity would have to be increased by 20 per cent. If the make-up is higher, the percentage of increase will be less. It will also be less with cleaner coal. Whether this is prohibitive or attractive will depend upon the rest of the circumstances.

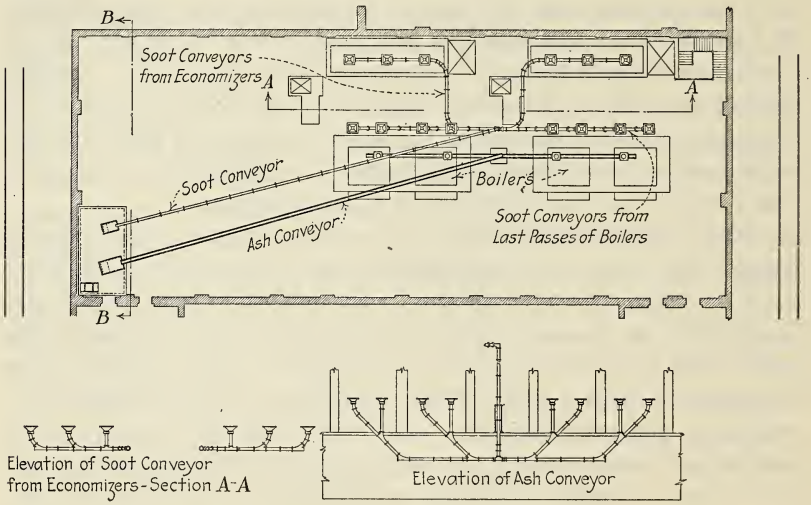
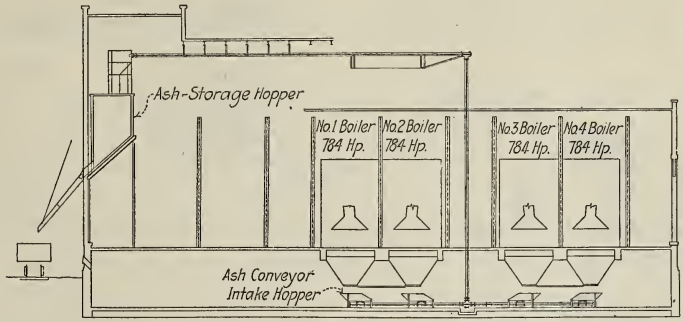


Fig. 24 Air Conveyor at Milwaukee Sewage-Disposal Power Plant. Plan

114 Air conveyors generally result in clean basements or firing floors because there is less spillage, as the tendency is always to draw the ash and dust into the conveyor. The first cost is usually lower than that of a mechanical system, especially when the only possible line of travel is tortuous. They take up very little space and can be installed in awkward positions, often such as are inaccessible to mechanical conveyors. The intakes and discharge can be arranged wherever desired, which is not always convenient with mechanical conveyors. They require very little attention; but this is not always a good feature unless periodical inspection is conscientious, because steam nozzles may wear and leaks due to abrasion

of pipes develop, and these may result in considerable waste of steam. The very convenience of the method may develop carelessness in allowing the steam to blow when ash is not being fed. It is safe to operate because there are no moving parts and this feature also makes for small expense of putting in renewals.

115 The air conveyors which are being installed by the Conveyor Corporation of America at the New Milwaukee sewage plant are illustrated in Figs. 24, 25 and 26. The plan in Fig. 24 shows the ash conveyor, which has a bore of 9 in., in conjunction with the four 734-hp. boilers equipped with Westinghouse underfeed stokers. The hopper ashpits are dumped into auxiliary hoppers formed about the conveyor intakes, as will be seen in Fig. 25. This permits of rapid feeding of the conveyor without



Elevation of Ash Conveyor

Fig. 25 Air Conveyor at Milwaukee Sewage-Disposal Plant. Elevation

smothering it. The elevation of the ash conveyor is shown in the lower right-hand view of Fig. 24. It is to convey a minimum of 12 tons per hour, with a steam consumption not exceeding 325 lb. of steam per ton of ashes.

116 The ashes are discharged into a target box above the storage bunker inside the boiler house. The storage bunker has a single slope bottom and is emptied through a movable chute into railroad cars as clearly shown in Figs. 24 and 25.

117 The conveyor for soot and fly ash is independent of the ash conveyor and has a bore of 6 in. This arrangement usually increases both the capacity and the efficiency over that obtained with combination conveyors. The arrangement of the soot conveyor is easily understood from Figs. 24 and 26. It is connected to the

later boiler passes and to the economizer, and discharges into a separate target box set upon the top of the ash bunker. As is seen in the left-hand view of Fig. 26, the conveyor pipes are connected directly to the boiler and economizer settings without the intervention of hoppers, and are provided with shut-off gates. Auxiliary hoppers are not necessary in conveyors for fly ash as there are no clinkers, and the conveyor can deal with this fine ash as fast as it can flow into the intakes.

118 The installation at the Lakeside plant of the Milwaukee Electric Railway and Light Company is of considerable interest

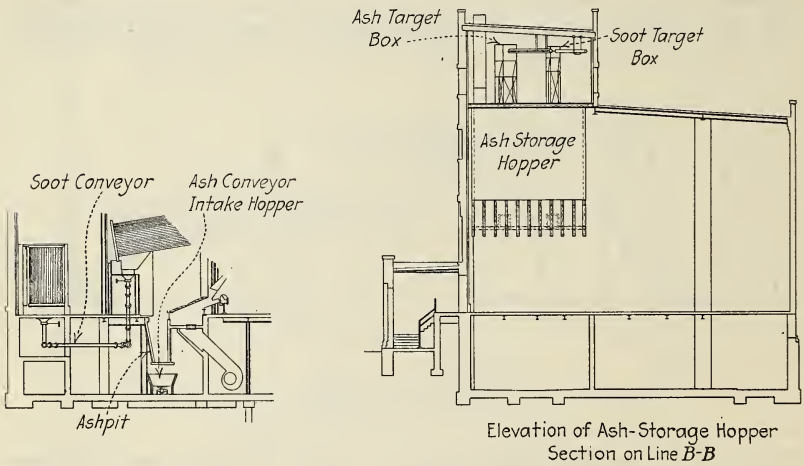


Fig. 26 Air Conveyor at Milwaukee Sewage-Disposal Power Plant. Cross-Section

because the boilers are fired with pulverized coal. There are eight boilers of 1333 hp. each — 1306 hp. in the boiler proper and 27 hp. in the form of a water screen at the bottom of the combustion chamber. This water screen extends over the entire combustion-chamber floor and might be called a tubular grid. This description is desirable because it appears that the water screen entirely prevents the formation of clinker or slag which was so troublesome in some of the earlier powdered-coal installations. Instead, the ash from the combustion chamber is a fine powder which is very easily handled by an air conveyor.

119 The air conveyors were installed by the Vacuum Ash and

Soot Conveyor Company and have operated in an entirely satisfactory manner from the start. They have been in operation nearly two years and practically no maintenance work has been required during that time. Only one nozzle has been replaced.

120 To recover some of the fine ash which would otherwise be discharged from the chimney top, a smoke washer is installed in the main flue about twenty feet before it enters the base of the stack. It consists of two 4-in. pipes placed parallel to each other in the top of the flue. These pipes are drilled and tapped to receive nozzles of $\frac{1}{4}$-in. pipe 1 in. long which are placed at 3-in. centers along each pipe, and the nozzles of one pipe are staggered relatively to those of the other. The pipes are supplied with about 350 gal. of water per min. at a pressure of about 6 lb. per sq. in. The jets from the nozzles form a water curtain through which the gases must pass before entering the chimney. Actual tests show that 50 per cent of the ash suspended in the flue gases is removed by this apparatus.

121 The water from the smoke washer falls into a trough formed in the bottom of the flue, and then runs off into a concrete settling basin carrying the ash it has collected. This basin is made of generous capacity to insure thorough settlement of the fine ash, and is 45 ft. long, 16 ft. wide, and 25 ft. deep. This roomy basin allows the inflowing water to become quiet so that the ash subsides; and it is then removed by a clamshell bucket operated by a locomotive crane. The water passes from the settling basin through overflow weirs into the circulating discharge tunnel from the condenser.

122 This smoke washer was installed after the building had been erected, and consequently cost a great deal more than if it had been designed and constructed in conjunction with the rest of the buildings and equipment. There has been practically no maintenance cost since it was started in operation.

123 The eight boilers are arranged in two rows of four each, facing each other across the firing aisle. There are two conveyors, the main line of each being in the floor of the ash alley in front of the furnace ashpits. Each main line takes care of four boilers. Branch lines lead to the ash chambers at the rear of the boilers and also to the ashpits under the economizers.

124 All the conveyor lines are of 8-in. cast-iron pipe. The running length of the two main lines is approximately 200 ft. with

a vertical rise of 65 ft. The lines from the ash chambers and the economizer ash hoppers are approximately 190 ft. long and have a vertical rise of 50 ft.

125 The conveyors discharge into ash bunkers, there being one bunker for each side of the boiler room. The ash from one bunker is discharged into cars, while the other is emptied down a closed chute into the ash settling basin of the smoke washer as

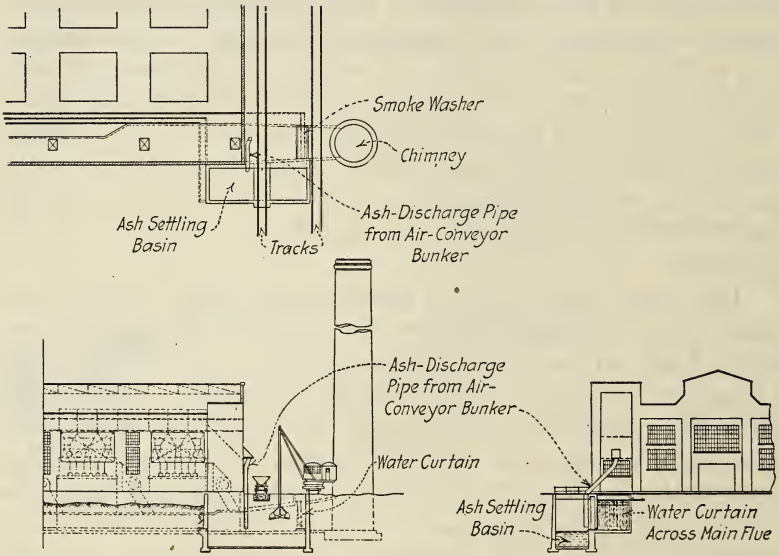


Fig. 27 Air Conveyor and Smoke Washer for Pulverized-Coal Firing at Lakeside Station, Milwaukee

shown in Fig. 27. As already mentioned, the ash from the settling basin is removed by grab bucket and put in cars to be hauled away.

ASH BUNKERS

126 Considerable choice of materials and design of ash bins and supports is available. A large number, perhaps most, are made of reinforced concrete. In some cases they are inside the boiler room and worked into the general design of coal bunkers, etc., such as illustrated in Fig. 20. They are occasionally built so as to span an alley, being supported by the buildings on each side.

127 The height of the discharge gate will naturally depend

upon what must pass under it and is greatest where standard railroad locomotives must be accommodated. Frequently, the discharge is at the side, the whole of the bottom sloping toward the door, which is then generally provided with a hinged chute, which can be extended so as to deliver the ash into trucks or railroad cars.

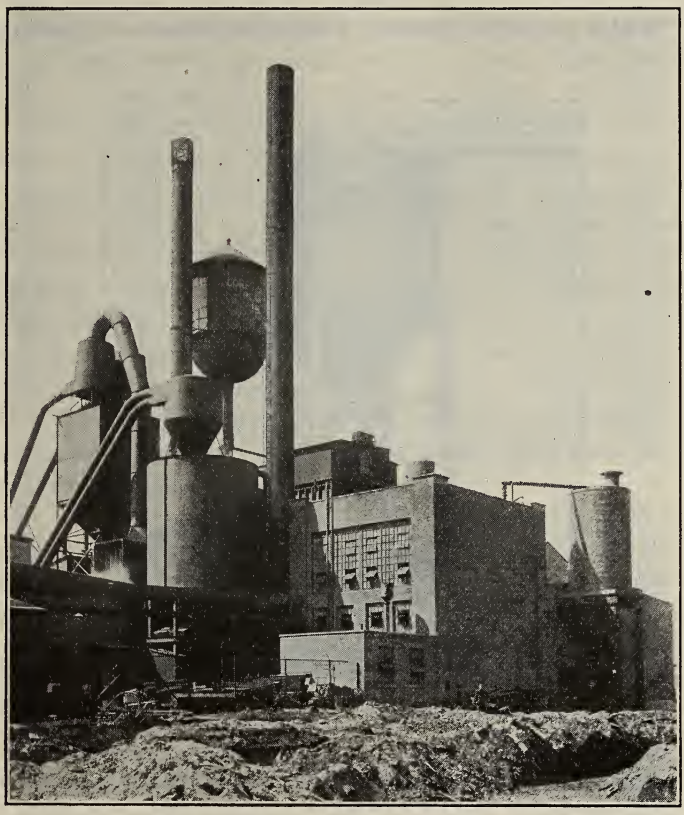


Fig. 28 Hollow-Tile Ash Bunker

128 Ash bunkers have also been built of brick, but unless very thick walls are used, they should either be buckstayed or have reinforcing bands laid up every five or six courses. The bricks should be hard and well burned, and laid in cement. The inner face of the walls should be of paving brick.

129 Fig. 28 illustrates one of the hollow-tile ash tanks in-

stalled for the Fisher Body Corporation by the United Conveyor Corporation. These tanks have several inherent advantages in that they are fireproof, prevent freezing of wet ash, are acidproof, provide space for steel reinforcing rings within the wall, and have a smooth interior surface. The tiles are laid up in cement and are provided with key grooves on their joint faces.

130 The Green Engineering Company install tanks which are made up of a cast-iron skeleton filled in with cast-iron plates, the

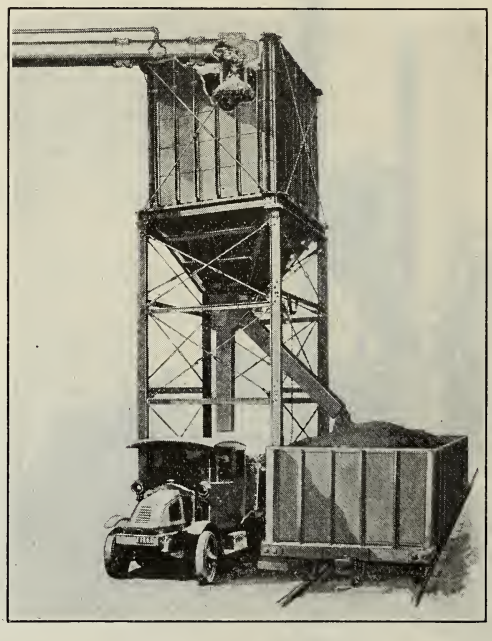


Fig. 29 Cast-Iron-Plate Ash Bunker

whole carried on a structural-steel framework as illustrated in Fig. 29. Cast iron, being much less subject to corrosion than steel, makes a durable tank for ash storage. The plates are easily removed and renewed. A substantial dumping door operated by a hand chain through gearing is provided. Poking holes are arranged around the dumping door so that bars may be introduced to break down any arching or choking of the ash.

131 Care should be taken in making repairs to storage hoppers to insure men working in fresh air, owing to the possibility

of their being "gassed" with carbon monoxide from combustible in the ash. This can occur in any case where closed or nearly closed storage hoppers are used. It is easily avoided with reasonable care.

WATER CONVEYORS

132 Systems of water conveyance as used on shipboard have already been described and were illustrated in Figs. 1 and 2. They consist of closed conduits carrying water and ash under pressure.

133 In land practice the conduit is largely or wholly open. Such open ducts are fairly common in mining and other plants, generally in hilly places where a brook may be diverted so as to flow along a flume in the boiler-room basement. Great care should be taken to be sure of a plentiful supply of water at all times. In one instance a flume was carried under the boiler room at one side of a narrow tunnel. The men stood back in the tunnel and hoed the ashes from a hopper ashpit into the flume. In comparatively dry weather and with red-hot ashes there was not sufficient water to quench properly and carry away the ashes. The heat, flame, and steam in the tunnel at such times were practically unbearable; and the labor cost of moving the ashes by dragging them along the flume by hand was very high.

134 With a plentiful supply of water, however, this method has much to recommend it. There is no dust or heat and the ashes are carried away very quickly. Owing to the low velocity of the vehicle as compared with air, the wear is very small and usually is not troublesome. The conveyance, of course, must always be down hill when open flumes are used, and a grade of 3 or even 4 per cent is desirable.

135 Where a natural supply of water from an elevation is not available a centrifugal pump may be used. The Hell Gate plant is an excellent instance of this kind, and the general layout is shown in Fig. 30. This is a system of open flumes within the boiler room continuing through a closed pipe to the ash settling tank. Tributary flumes are carried along below each line of boilers and empty into a main cross-flume which runs along the boiler room wall near the turbine room. This main flume then turns and becomes a full pipe or enclosed conduit leading to a pit near the river side into which it discharges. The ashes are recovered from this pit by a grab bucket operated by a locomotive crane running on

a track laid on the pier, and discharged into scows. The scows are towed out and dumped at sea about five miles east of Sandy Hook, or about thirty miles from the plant.

136 The flumes within the boiler room are of concrete with a bottom lining of vitrified earthen drain tiles or half-pipes. Those under the boilers are supported on structural-steel framework suspended from the firing floor, while the main cross-flume is carried on steel trestling.

137 A cross-section of the flume under the boilers is shown

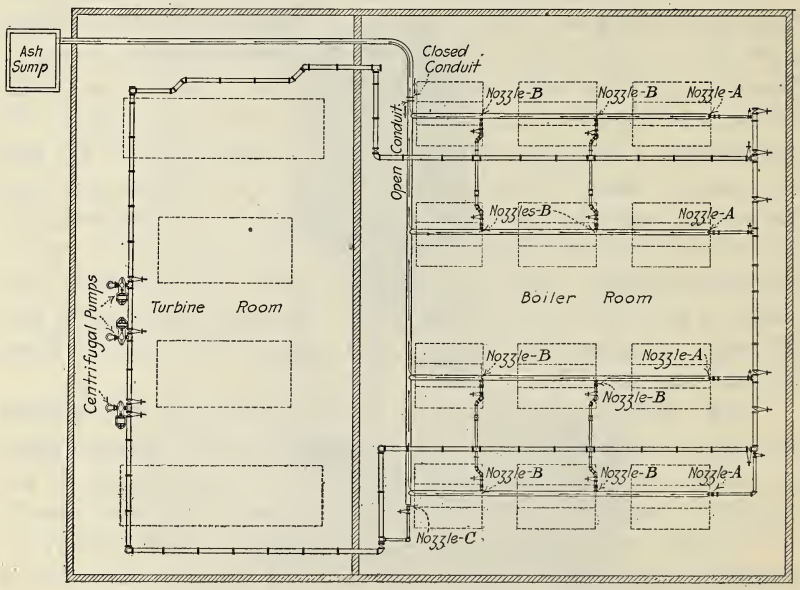


Fig. 30 Water Conveyor at Hell Gate Station. Plan

in Fig. 31. The ash as it leaves the clinker grinders drops directly into the water and is carried away. Access doors lined with common brick are provided so that any obstruction can be handled easily. These doors may be seen in Figs. 31 and 32. Fig. 32 is a longitudinal view of one of the ducts under the boilers.

138 The construction of what may be termed the ash chute, there being no ashpit proper, may be compared with some interest with the unlined sheet-steel chute at Poplar as shown in Fig. 22.

139 The method of water supply is very interesting. There is a nozzle *A* at the head of each tributary flume, and an ingeniously

arranged undercurrent nozzle *B* at the beginning of each succeeding ashpit. These undercurrent nozzles are at the bottom of the flume and are arranged to discharge horizontally downstream. They are placed to form steps in the flume so that the ash flows over them. They are shown a little more clearly in Fig. 33. At the head of the main conduit is a booster nozzle *C*.

140 The water for the various nozzles is taken from the

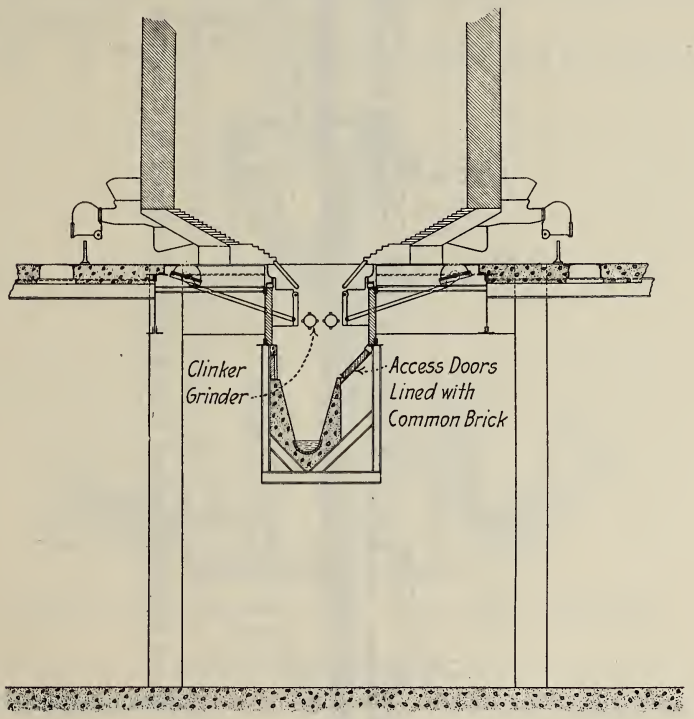


Fig. 31 Water Conveyor at Hell Gate Station. Cross-Sectional Elevation

circulating discharge tunnel and is supplied under pressure by 12-in. Lea-Courtenay centrifugal pumps direct-driven by Westinghouse motors. These pumps are single-stage double-suction volute pumps and supply 5000 gal. per min. against a head of 75 ft., operating at 81.5 per cent efficiency. The motors are rated at 150 hp., but the pumps require only 125 hp. under regular operating conditions.

141 The main pipe line which supplies the water nozzles has a bore of 16 in. and is arranged as a ring, as clearly shown in Fig. 30.

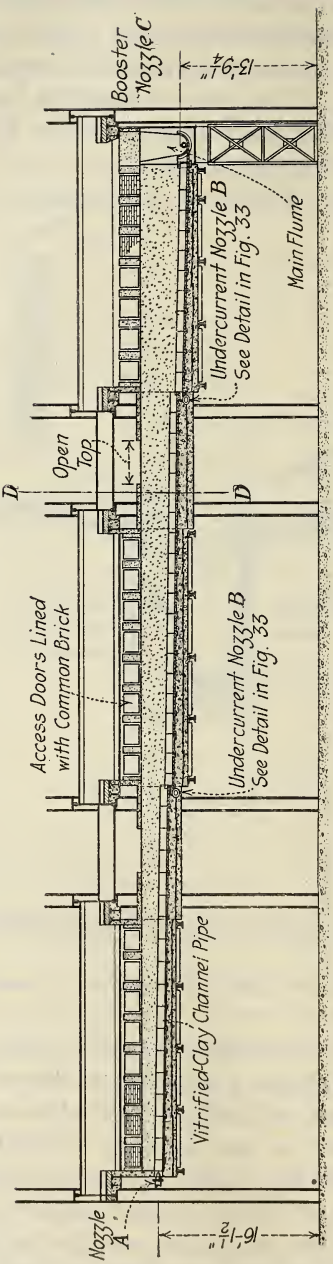
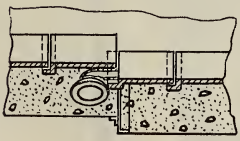


Fig. 32 Water Conveyor at Hell Gate Station. Elevation of Tributary Flume

Gate valves are placed in such positions that either side of the ring can be shut off for repairs. The three pumps are connected to the ring, with valves so disposed that any or all of the pumps may supply either side of the ring main and any one or two of the pumps may be isolated for overhauling without interrupting the supply of water. The valves controlling the supply of water to the individual nozzles are located in the basement convenient to the flumes, so that adjustment of the flow of water along the flumes can be made very easily and quickly.

142 The system has not been sufficiently long in operation to

Section on Line *D-D*



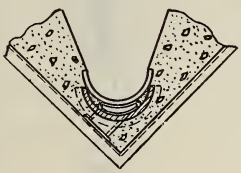


Fig. 33 Water Conveyor at Hell Gate Station. Detail of Nozzles and Flumes

determine maintenance costs. The only operating charges are the wages of the ash-crane operator and the cost of energy for driving the pumps. Either one or two pumps are operated according to load and quality of coal.

143 It will be noticed that these ashpits are not sealed, but with the amount of ash lying on the clinker grinders in usual operation, any serious excess air is not to be expected with the forced-draft stokers used.

144 Fig. 34 shows the general arrangement of the ash-sluicing system of the Lacombe station of the Denver Gas and Electric Light Company, and a cross-section of the flume under the boilers

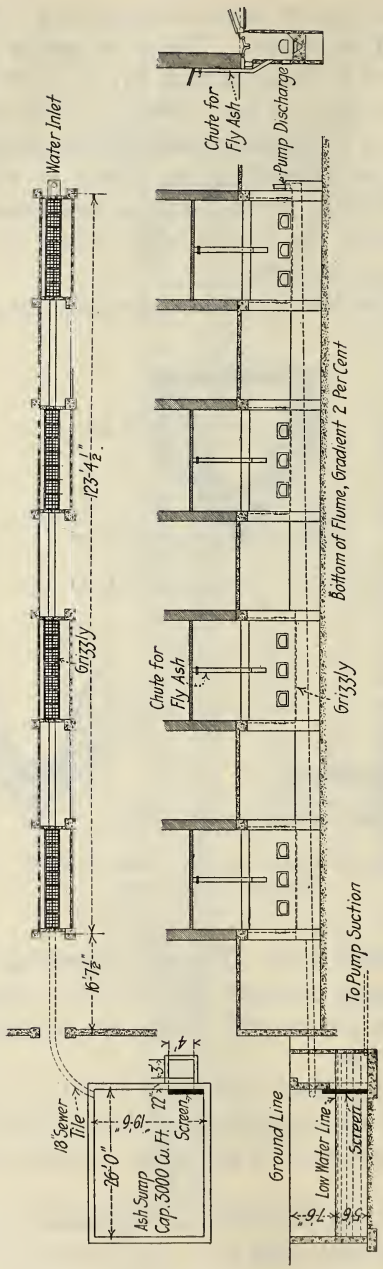


FIG. 34 WATER CONVEYOR AT LACOMBE STATION, DENVER

is shown in Fig 35. The outstanding difference between this plant and the one at the Hell Gate station is that there are no clinker grinders. Consequently, instead of the ground ash dropping into the flume continuously, the stokers are dumped at intervals, sending down a considerable bulk of ash containing large clinkers.

145 As a result, the flumes are protected by a "grizzly" composed of heavy bars set 6 in. center to center to withstand breaking up the large clinkers until they can drop between the

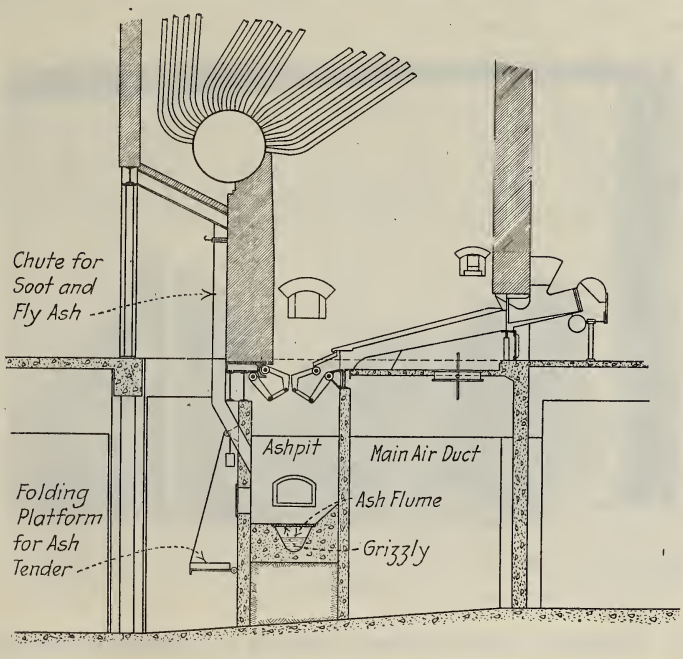


Fig. 35 Water Conveyor at Lacombe Station, Denver. Cross-Sectional Elevation

bars into the flume. Extremely large clinkers would stop up the flume, but it is interesting to observe Fig. 36, which illustrates some clinkers which have actually passed through the flumes. The water stream will easily handle clinkers about 12 in. square and 15 lb. in weight.

146 The grizzly is made in sections, there being seven sections, each 30 in. wide, to each boiler. Any or all of these sections can easily be removed to give access to the flume.

147 Water sprays cool the clinkers which are caught on the grizzly. The water sprays are discharged from 2-in. pipes which run the full length of the ash hoppers, and which are drilled with $\frac{1}{4}$-in. holes on 3-in. centers. As the holes of one pipe are staggered relatively to those of the other, a spray of water is directed across the hopper every $1\frac{1}{2}$ in.

148 Side-hinged doors 18 in. by 23 in. are provided through which the clinkers can be managed. There are three doors at the rear of the flume and one door at each end under each boiler. These

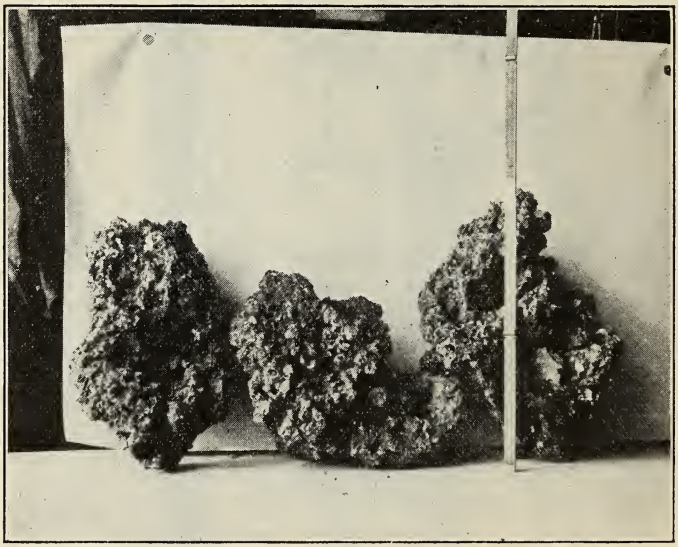


Fig. 36 Clinkers Carried by Water Conveyor at Lacombe Station, Denver

doors may be seen in the figures. It is wise to allow ample access to the flume to remove any obstruction, such for instance as might possibly be caused by ashes being inadvertently dumped while there was no water flowing, or by a fall of firebrick, since firebrick cannot be sluiced.

149 The station contains five boilers of 750 hp. each, operated at 250 to 300 per cent of rating. The coal is a sub-bituminous containing 6 to 7 per cent of ash. As the boilers are in line, the flume is single and straight. It is all of 2 per cent grade except the curve to the ash-settling tank, which is of 18-in. vitrified sewer

pipe laid horizontally. The flume and sewer pipe is about 155 ft. total length. The capacity of the settling tank is 3000 cu. ft.

150 The water passes through a screen and is recirculated, and a 2-in. line is used to replenish occasionally the water in the system. The water is circulated by a 6-in. American open-runner centrifugal pump driven by a 20-hp. motor. The pump delivers about 1100 gal. per min. against a head of 25 ft. The system deals with about 33 tons of ashes in 24 hours.

151 The stokers are dumped at intervals of two to two and one-half hours. Water does not flow continuously. When the fireman is about to dump ashes the pump is started, and when the

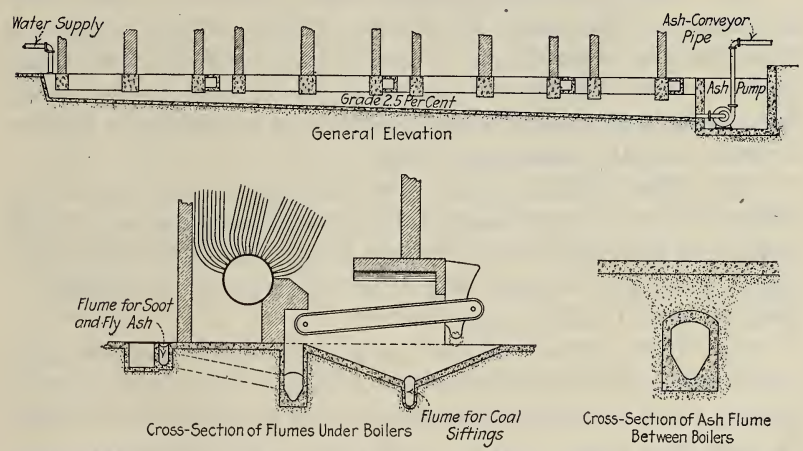


Fig. 37 Morrison's Water Conveyor at Great Western Sugar Refinery, Denver

sluiceway is filled with water the ashes are dumped from one boiler at a time into the swiftly moving water. A whistle signal is in use between the fireman and the ash tender. The ashes are recovered from the settling tank by grab bucket and given to the city for suburban road making. Railroad facilities are also provided so that cars can easily be loaded by the grab bucket.

152 Three combination men, one on each shift, are required to handle the ashes and attend to the feed pumps; and about one-third of each man's time is devoted to ash handling. Two men, one on each of two shifts, operate the crane and clamshell bucket by which the ash is recovered from the settling tank. Each man spends about one hour moving ashes.

153 An ingenious arrangement is that invented and patented by Mr. Morrison, of the Great Western Sugar Company of Denver, and installed in several of their plants. An example is partly illustrated in Fig. 37. A flume into which the ashes are fed is arranged under the boilers as in other water conveyors. This flume carries the ash-laden water into the suction connection of a centrifugal pump which discharges into the conveyor pipe. The conveyor pipe is of cast iron, 6-in. bore. Abrasion occurs slowly, but is confined to the bottom of the pipe. It is found that by rotating the conveyor pipes through an angle of 90 deg. every few years, maximum service is obtained out of each renewal.

154 The most serious expense would seem to be that caused by abrasion of the pump; but it has a manganese-steel lining, and the authors are assured that renewal of this lining and of the pump shaft are only necessary at long intervals. These installations are highly satisfactory, and the ease with which the ashes are handled far outweighs the maintenance cost.

155 The discharge pipe from the pump should not slope downward, as it is then found that the water has a tendency to run away from the cinders which remain in the pipe and cause a stoppage. When the discharge pipe is laid horizontally or sloping upward, no such trouble occurs.

156 In the installations so far made the ashes are discharged into natural depressions near the plants. There are plentiful natural water supplies so that there is no necessity to recirculate the conveying water. But it would appear that the discharge pipe could be led into an elevated hopper which would act as a settling tank. The clear water from an overflow could then be carried back to the head of the ash flume. When the pump was stopped and the tank drained, it could be emptied into railroad cars or motor trucks. Such a tank should probably be equipped with a fairly large dumping door as the ash would tend to pack rather tightly.

157 In designing a water conveyor it must be borne in mind that while continuous dumping stokers such as chain grates or those equipped with clinker grinders may discharge directly into continuously running water, dumping stokers or any firing system where large clinkers are to be expected should discharge on to a grizzly of massive bars on which clinkers may be broken small. The water need only be running while dumping is in progress with stokers of the dumping type.

158 Concrete troughs, if carefully built, will be found to require very little repair. The inside lining should be smooth and free from pits.

159 There should be a plentiful supply of water under a few pounds pressure. About 1000 gal. per min. should be supplied for each flume.

160 With natural draft, means should be devised to provide seals so as to prevent too great an excess of air from passing up through stoker dumps.

161 When the water is recirculated, the pump should be designed to handle gritty water, and occasional renewals due to this reason must be expected, though they will not be serious.

162 As there is often considerable dust and gas generated with stokers of the dumping type, it is advisable to provide the access doors with latches so that they cannot be blown open. Ample ventilation near where ashes are handled should always be provided.

163 While but little attention is necessary with this method of handling ashes, it is well to have a man told off for the purpose. One man can attend and clean the circulating water screens, supervise the sluicing of ashes, and generally act as boiler-room janitor.

164 In some localities in winter, trouble may be expected from water freezing in the bearings of grab buckets. In particularly cold situations the wet ash will freeze in the cars and cannot be dumped unless it is thawed out. Weather conditions may therefore prohibit water conveyors in some instances.

ASH DISPOSAL

165 As mentioned when describing methods of conveyance, ashes are often used to fill depressions around the plant, but this cannot go on forever and means must be adopted to get them away from the vicinity. Some highly fortunate persons are able to sell them, but most are glad to be able to give them away. It is of great advantage in both of these methods if the ashes are loaded into an overhead bunker of some kind which will discharge by gravity through dumping doors into carts and motor trucks. People taking the ashes away prefer to patronize those plants where their trucks can be filled quickly in this way rather than to spend the time and money to shovel them from a heap on the ground.

166 But the ashes must be got away somehow, and if they

cannot either be sold or given away, they must be removed by rail or by boat. They must not be dumped into rivers, partly because of pollution and partly because of choking the river bed against water flow and against navigation. If removed by rail, they are usually dumped finally into natural or artificial depressions, etc., where filling in is desired or tolerated, such as ravines and abandoned mines.

167 At some of the plants of the West Penn Power Company it is intended to carry away the ash by aerial tramways or ropeways as far as 3000 ft., where it will be used to fill up ravines to a height of from 200 to 300 ft. At the Windsor plant, where ashpits similar to those of Springdale as shown in Fig. 16 are to be installed the bucket will dump the ashes into a hopper feeding a belt conveyor which will discharge into an outside tank from which the aerial-tramway cars will be loaded. At the Springdale plant the ashes will be loaded from the bucket into a traveling hopper which will discharge into the aerial cars without any connecting link.

168 Some public-utility companies have a number of isolated plants used for the supply and sale of steam for heating, which take all their electrical requirement from the central station. In one such instance in the Middle West the ashes are collected in trucks and wagons and hauled to a point on a railroad siding. An elevated ash bunker having a capacity of two railroad cars was erected here at a sufficient height to discharge ashes into the cars. A bucket elevator was installed at the side of the bunker, with the boot below the ground level. A grating was placed level with the ground and over a hopper or chute leading into the elevator boot, and the trucks and wagons drive over this grating and dump the ashes through it. This is a very convenient method of disposing of ashes from a number of small plants as no delays are encountered by wagons waiting to unload.

169 Plants near the coast can sometimes dispose of their ashes by loading them into dump-bottom barges which are towed out to sea for a prescribed distance, where they may be discharged. This method is practiced by the Hell Gate plant as already described.

170 The ashes are sometimes used for making suburban roads. When in small enough pieces such as will pass a $\frac{1}{2}$-in. or $\frac{3}{4}$-in. screen, they are often used for making cinder concrete.

171 In considering any new station, the ultimate disposal of

the ashes is one of the first things to be considered, because the ash-handling equipment can then be devised so as to accommodate itself thereto in the simplest manner.

CONCLUSION

172 Costs of equipment, maintenance, or labor have not been included, although numerous cost figures have been collected, some giving very exhaustive details of cost of installation and operation. Such figures are of absolutely no value for comparison as between one station and another, or as between one system and another, and are likely to do more harm than good. So many circumstances enter into any particular problem that any figures offered for comparative purposes should be used with great caution. Wide experience and sound judgment are far more valuable in this work than bald figures, and therefore no tabulation of costs of any kind has been attempted.

173 The primary object of any modern system of ash handling is to reduce labor costs to a minimum. Any scheme must conform to this requirement; and then must conform to the further requirements of low depreciation and maintenance.

174 The method of ash handling adopted for any plant will be governed by the way in which the ashes are delivered from the fire and by the facilities for final disposition. It may even be the case that a system must be so devised as to meet efficiently an intended change at either or both ends. Reasons may exist compelling the method of firing to be changed some time after the ash-handling system has been installed, and the same thing may apply to the final disposition of the ashes. The prudent engineer will examine into future as well as present requirements.

175 The endeavor has been made to present sufficient description and illustration of all the successful methods of handling ash in such a manner as will enable those interested in the subject intelligently to design or criticise almost any installation with which they may be confronted.

176 In conclusion, the authors desire to express their grateful acknowledgment of the ready and courteous response of those connected with both manufacture and operation when requests for information were made.

DISCUSSION

T. A. Marsh. For a long time, power plant designers have felt the need of a comprehensive treatise on ash removal and disposal. Power plants are frequently successes or failures, depending upon whether a good ash disposal system or a poor one has been installed. This refers not only to the type of equipment for handling the ashes, but the design of the ashpits or hoppers, their capacity, the valves, the quenching system, and in fact all of the details that go to make up the success or failure of the system.

Designing engineers and consulting engineers, building power plants frequently, have had some basis to govern their designs, but the occasional builder of power plants, such as the average industrial company, has been without any text or data to assist in this important feature of power plant design.

From the standpoint of the stoker manufacturer, ashpit design is extremely important, as faulty ashpits have contributed more to the failure of stokers than any other single item, except, perhaps, draft.

Par. 82 and those following describe a type of cross conveyor used extensively in England and gaining great popularity in this country. These conveyors can be installed where the head room is limited. They are built very ruggedly and are producing wonderful results. In the large central stations, the application of this type of conveyor permits the reduction of head room in the basement, thereby saving materially on initial cost, and on operating heights for elevating coal and other material.

Referring to Par. 103, on air conveyors, it is a question as to whether it is desirable to use conveyors larger than 8 in. in diameter. The average capacity of a man hoeing ashes to an intake is five to six tons per hour. Eight-inch systems will handle seven or eight tons per hour, but are invariably limited by the ability of the operator to get the ashes to the intake. Therefore, increasing the size of the conveyor pipe does not proportionately increase the capacity of the ash-handling system.

Remote control, mentioned in Par. 108, for shutting off steam when the system is out of use, is being received with great favor. In the average system about twice as much steam is used as is actually required for handling the ash, the excess steam being

that which is consumed while ashes are not actually hoed into the intake.

Another very important item in air system design and operation is the wear of steam nozzles. So far no metal has been found that will resist this wear for a long time. A very slight increase in the diameter of a nozzle, of, say, $\frac{1}{4}$ or $\frac{3}{8}$ in. diameter, increases the steam consumption enormously. This makes the steam consumption disproportionate to the ash handled, and creates too high a velocity throughout the system, thereby causing excessive repairs and usually discrediting the ash system from the basis of poor steam economy and high maintenance.

The thin shell-like nozzle is the safest and surest design to prevent these troubles, for it is impossible to increase the steam consumption with this. The nozzle simply wears through and can be quickly replaced. The cost of such nozzles is but a small fraction of the cost of the solid hardened metal design.

F. B. Allen. The ordinary system of quenching ashes is to introduce one or two pipes, perforated with a series of holes, running the entire length of the hopper. Water is then connected to these "spray pipes" and squirts out of them in a series of jets. The most serious objection to such a system is its failure satisfactorily to quench the ash, unless a tremendous excess of water is used. An inspection of such a system in operation clearly indicates the reason. Each jet wears a small channel through the ash to the gate beneath. The water cools the walls of its particular channel, leaving the rest of the ash unquenched. This is strikingly checked by feeling of the water as discharged from a hopper thus equipped, which will be found to be lukewarm; if then the inside of the hopper is inspected, it will be seen that a great portion of the ash is still incandescent.

Much better results may be obtained by running the quenching water header outside of the hopper with a series of off-takes to quenching nozzles located through the hopper walls. These quenchers are designed to spray the water in fine particles evenly over the ash. There will then be no channels worn through — each drop of water will cool its quota of ash.

Another serious objection to the use of perforated pipes is the clogging of the perforations. As a result, the water distribution is, of course, even poorer and often the quenching system is virtually shut down. When these perforations do clog, nothing can ordinarily

be done about cleaning them until the boiler is shut down and the pipe can be cleaned from the inside of the hopper. Another objection is the inability of the operator to control the water quantity entering the hopper, for the reason that plant pressure is ordinarily connected to the spray pipe, the water supply being controlled by a valve. If the line is throttled by this valve, the pressure of the pipe is reduced and the water merely dribbles from the opening. A system of spray pipes within the hopper is subject to rapid deterioration resulting from their constant contact with hot sulphurous gases.

Where a series of quenching nozzles are used, each independently controlled, the water discharge from the quenchers may be controlled over a wide range by throttling, with little effect on the distribution, or alternate quenchers may be entirely shut down, the others giving very satisfactory water distribution for lighter loads.

It may be argued that these quenching nozzles will clog, making them ineffective. Such trouble is very simply met by connecting each quencher through a $\frac{1}{2}$-in. valve to supply header. Thus any quencher may be shut down. The quencher is so designed as to have a minimum orifice of $\frac{1}{4}$-in. diameter and should this clog, it is possible to clean it from the outside of the hopper by removing a 3-in. acid bronze access plug. This can be done without disturbing the operation of any other quencher and while the boiler is in operation.

With a series of quenchers located in the walls of the hopper, the quenching header is of course outside of the hopper and is therefore not subject to the deteriorating effects of contact with hot sulphurous gases.

It is the writer's opinion that ashes should be continuously quenched. Probably this opinion will not meet with much argument where the ashes are dumped continuously, as is the case with chain-grate stokers, or underfeed stokers with clinker grinders. In such cases, the ashes are discharged to the hopper continuously and it is logical to quench them continuously. Where the ashes are discharged intermittently from dump plates, particularly in connection with underfeed stokers, it is probably advantageous to reduce the amount of continuous quenching water perhaps 50 per cent; using the additional 50 per cent of water for only a comparatively short time after each dump of the stoker.

Continuous quenching with the water introduced by means of nozzles requires but from 3 to 5 gallons per minute per hundred rated horsepower of the boiler; the pumping expense is therefore

small, while the advantages are considerable. If the ash gates beneath the hoppers are designed with a very small amount of clearance between the frame and the movable door, the quenching water will build up a slight head in order to force its way through this opening. Manifestly, if there is 1 or 2 in. of water on the top of the gate, no air can enter the hopper around this opening and the hopper is water sealed. The arrangement results in continuously cool ashes, meaning a minimum up-keep of stoker parts at the dump end. It also means a maximum combustion efficiency as there is no excess air leaking into the furnace from the ash hopper which is water sealed.

A wet and sloppy ash aisle beneath is obviated by use of an ash gate which collects this water as it flows from the hopper and discharges it to drain, thereby maintaining a clean, dry basement.

Nevin E. Funk. Ash-handling systems can be divided into two divisions. One class is easy to operate, may be readily repaired in parts without putting the entire system out of commission, and if it does fail some other device can be substituted for it without putting the boiler plant out of service. Lastly it is flexible and can be used for greater capacities than originally intended. The apparatus of the other class is the antithesis of the first. It is not flexible, when repairs are necessary at least one section of the boiler room is out of commission, other methods cannot be substituted for it, and it is generally of greater initial expense and may be more expensive to maintain.

In the first of these two classes the writer would include the railroad car, the industrial railway with storage battery or trolley locomotive, and perhaps the water-sealed ashpit of Fig. 16 with a crane for removing the ashes. In the second class should be included all air, hydraulic and mechanical conveyors.

The difficulty of raising the boiler room to accommodate railway cars is not so serious. There is not much more room required with such a system than with some others, as it is possible to drop the basement where the cars come in.

The author says in Par. 26 that the "largest ashpit capacity will be requisite with conveying systems that are operated intermittently." This is predicated on the fact that the continuous system operates all the time, which it does not. There must be sufficient storage to provide for the intermittent operation caused by breakage.

The system illustrated in Figs. 20 and 21 is for both coal and ash. Coal has enough ash in it without the addition of ashes which may drop into it from the buckets of this system.

The water-seal chain and cross-bar conveyor of Fig. 22 does not look promising from an operating standpoint. It does not seem possible that one chain would break without breaking the one beside it, and with a boiler in operation it would be hard to repair the conveyor.

The air conveyor with vacuum storage tank, illustrated in Fig. 23, makes a good water gas generator which is likely to explode. Also the system is a hard one to keep tight, and the wearing backs are not the only parts that wear.

The hydraulic system of Fig. 30 looks as though it would put three boilers out of commission in event of failure, and while the flotation of ashes by water relieves some of the weight so far as friction is concerned, there is still abrasive action.

With the railroad car or industrial railroad, more cars can be used if the ash content of the coal increases. The cars can be pulled out by horses in case of breakdown, and if the rails break it is possible to send trucks into the basement to carry the ashes away.

Nixon W. Elmer. A good many engineers feel that we are entering an age of powdered fuel. The powdered-fuel ash problem is different, and requires considerable study.

I. E. Moultrop. Of the three big handling problems of the modern power plant, — water, coal and ashes — that of the ashes is the most perplexing. But the ash problem of a big plant is not the handling but the disposal.

The first problem of ash handling is to provide a big ashpit. It cannot be too big. Ashes should be handled with the least amount of machinery possible, and the machinery should be such that, in case of breakdown, it does not interfere with the disposal of the ashes. The railroad or the industrial railway have the advantage that in case of breakdown the ashes can be dumped on the floor.

T. Maynz. The problem of disposing of ashes in a large station is a difficult one, and it is important to have a space sufficient to store the material. In our plant there are industrial cars and a skip hoist. In case of a breakdown, the ashes are dumped on the floor which fortunately will accommodate 48 hours' supply of ashes.

The problem is not to cut down on head room in ashpits but to give them all the room possible.

Sam H. Libby. One of the greatest problems of a designing engineer is to find out all the details of what is expected of the machine he is designing. Recently a machine was supplied to a concern who said that their coal capacity was 20 to 40 tons of coal per day. The machine was designed for this capacity, with allowance for an ample margin but it broke down. Investigation showed that the coal was bought in train-load lots arriving at the plant in lots of six or eight cars at a time, and unloaded as rapidly as possible to save demurrage. The machine built to handle 40 tons per day was consequently handling 250 tons.

E. H. Tenney. The disposal of ashes from numerous isolated power plants, such as office buildings, hotels, etc., in the down-town district of the city of St. Louis has had considerable attention by the writer. Practically all types of systems are in use, each being adapted to its own particular requirements, such as, wheel barrow, bucket hoist, hydraulic lift, motor-driven lift, air ejection system and the link-belt system described by the author in Par. 70.

In connection with the use of air ejector systems in down-town districts, we found, that other things being equal, a great deal of consideration had to be given to the prevention of dust. In the particular installation in mind, this was eliminated by the careful design of water spray rings and the proper installation of a standpipe vent on the ash receiving tank. We have also found that where ash hoppers are located outside the plant so that trucks or other conveyances can drive beneath them, for loading purposes, considerable difficulty was encountered in the winter, due to the ash gates freezing. This difficulty was easily overcome by the installation of small steam lines on all such ash-hopper gates for the purpose of thawing them loose. This point should not be lost sight of in building overhead ash hoppers where the question of hurried ash removal is necessary, as is the case with office building or hotel power plants.

The writer agrees with the authors as to their conclusion in that each handling problem must be given its own due consideration and that what is good for one station cannot be acclaimed good for another, and for this reason the presentation of the means and methods employed by others, as set forth in the paper, readily

visualizes the many schemes possible, and no doubt should prove very helpful to those considering ash-handling problems.

Rankin Eastin[1] wrote a description of an hydraulic ash-disposal system installed by him at the Tell City Water and Light Co. An 18 by 18-in. sluiceway was constructed of concrete beneath two boilers (one 406-hp. Heine and one 250-hp. Atlas) in such a way that the ashes fall into it and are carried out onto a catch platform where they are separated from the sluice water and stored for use. In case there is no demand for the ashes they are allowed to discharge into the river with the sluice water. Water for the system is derived from the waste water from condensers. Emergency water connections with water at 70 lb. pressure are also provided. Eight-inch connections to the sluice are provided for soot and fly ash from the rear of the boilers.

R. H. Beaumont. The writer's conception of a modern ash handling system is that it must be capable of meeting the following requirements:

1 It must be capable of handling very large clinkers
2 It must be of such design that abrasion has little or no effect on its parts, so that repairs are negligible
3 It must not consume much power, nor waste it running idle
4 It must be capable of lifting the ashes to a good height without being subject to excessive wear by so doing
5 It must handle either red hot or dripping wet ashes with equal facility
6 It must operate without making dust or exceptional noise
7 It must be capable of handling ashes at a high hourly capacity
8 It must handle the general run of boiler room refuse, such as soot, flue dust, broken bricks, broken grates, etc.
9 It must be capable of taking care of plant extension without having to be completely remodeled
10 It must deliver ashes into the storage bunker in such a condition that they are acceptable to the railroad as freight.

The writer was disappointed with the casual mention of what is the most important device used today to handle ashes —

[1] Supt., Tell City Water & Light Co., Tell City, Ind.

the automatic electric skip hoist. However, while but slight space is given to it, it is pleasing to note that it is a very satisfactory device, and no faults are to be found with it. In fact it is the only popular device described in the paper which seems to get a clean bill of health.

The fundamental idea is covered by Par. 173 wherein it states that "the primary object of any modern system of ash handling is to reduce labor costs to a minimum." The ash hoppers are part of the system and should be proportioned with this fact in mind. In general they should have enough capacity so that all the ashes produced in 24 hours can be easily handled by the day shift, which means somewhat over 16 hours and preferably 20 hours storage based on a safe percentage of ash content.

The writer wishes to modify the statement made in Par. 26 that where a continuous conveyor system is used, the hopper may act merely as a funnel or chute to guide the ashes into the conveyor. Of course, it may, but this means that ashes must be handled continuously every shift and also that no time is available to repair the conveyor. The hopper must still be proportioned to save labor if the fundamental reason for installing the system is not to be violated.

Continuous conveyors for ashes cause complete paralysis when they break down.

Care must be taken to design the hopper to fit the system. For instance, bottom dump hoppers with large gates five feet square will frequently empty the hopper too quickly. This would not be objectionable if the hopper were discharging into a railroad car or large truck, or any other container that would take the entire contents. In this case the gate is not used to control the flow of ashes, but rather to close the bottom against the time of discharge. But suppose the hopper, holding 20 hours' production of ashes, discharges into a small industrial car. It then becomes very necessary to control the rate of flow in order to prevent flooding the car.

A design of hopper with gravity discharge and at the same time having perfect control over the flow of ashes is shown in Fig. 38 where the outlet gate is set at an angle of 45 deg. which is also the angle of slope the bottom plate.

Par. 35 states that steel-plate hoppers are undesirable owing to rapid corrosion and therefore it is best to use cast iron. Steel-plate hoppers have been in service many years, while the cast-iron hopper is a very recent innovation and consequently it is impossible

to state with any assurance that it is better. The writer believes that plate hoppers, $\frac{3}{8}$ in. thick, made of pure iron and protected with bitumastic enamel between the plate and the brick lining are better than cast iron. Hard burned or vitrified paving brick is better than firebrick for lining the hoppers.

Cast-iron hoppers are supported by a steel framework, which

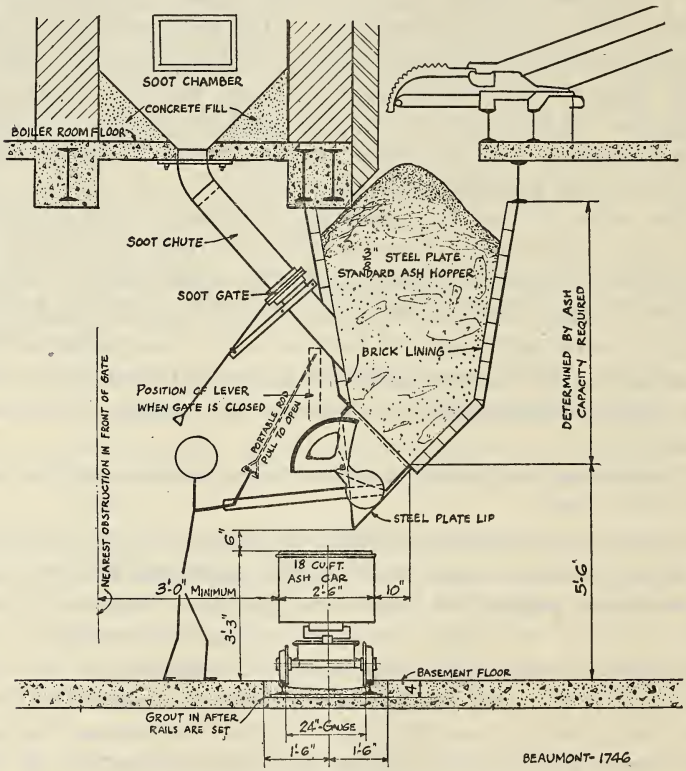


Fig. 38 Hopper With Gravity Discharge

in turn is attached to the boiler beams by suitable connections. In comparing them with steel plate hoppers, it must be remembered that the horizontal sectional area of these connections is very small whereas the steel-plate hopper is continuously supported all around. In other words, how much supporting metal must be corroded in either case before the hopper can drop to the floor.

Referring to the push cars which are shown in a number of the illustrations, my experience has been that the most one man

can push is 20 cu. ft. Wherever a larger push car is used, two men will be found pushing it even though the designer of the system intended it to be a one-man car.

The modern tendency is to use an electric ash larry of 50 cu. ft. capacity, designed with ample power to haul about three additional trailers each of 50 cu. ft. which may be added one at a time as the plant grows. These electric ash larries with their trailers are bottom dumping and designed so that the gate trips automatically when passing over the dumping point. A representative layout of such an ash larry and trailers is shown by Fig. 39.

The writer believes that the ash larry, used in connection with an automatic skip hoist, the entire system being operated by one man, is today the most widely used ash-handling method, for medium and large-sized boiler houses because it better meets the factors which must be met to secure a good ash-handling system, as mentioned in the first paragraph of this discussion. No conveyor manufacturer fostered it. In fact, it was developed and applied to a Philadelphia Rapid Transit Co. power house in 1897, to replace the rather crude chain elevators and conveyors then in use. Another installation made about 1908 was at the Lardners Point Pumping Station of the Philadelphia Water Works which is still in service. Some of the very oldest stations of the New York Edison Co. used skip hoists.

The question of large clinkers is a serious one and is recognized as such in the paper under discussion, but the problem is not to be solved by merely providing something to prevent clinkers entering the system. Of what use is it to put gates five feet square under the ash hoppers in order to pass these clinkers and then make the remainder of the system capable of handling only fine ashes.

Mention is made in Par. 118 of the justly famous Lakeside plant of the Milwaukee Elec. Ry. & Light Co. which uses an air conveyor. It is interesting to note that a year or so ago a skip hoist was installed to handle the clinker or slag.

It is not stating the case correctly to say that there are no moving parts to an air conveyor of the steam jet type. In this case, it is entirely justifiable to say that the entire system is one immense moving part, as movement is relative, and it is certain that the ashes are rushing at a great rate through the pipe. The relative movement between the pipe and the ashes certainly ought to be taken into account. With a car and skip hoist system there is no relative movement between the car or skip bucket except when

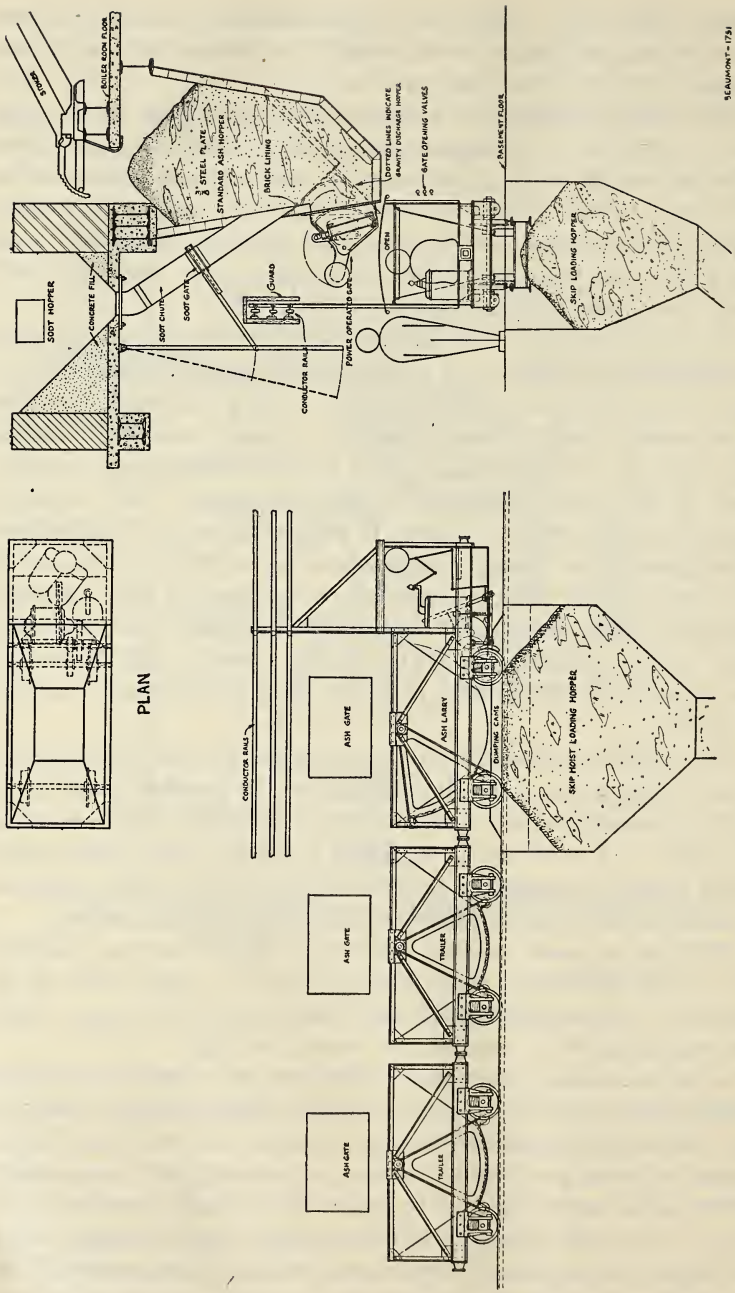


Fig. 39 Layout of Ash Larry and Trailer

loading and discharging and even then, the amount of ash in sliding contact with the plates is but a small percentage of the total amount contained within the bucket. During the transfer from hopper to bunker the ashes are absolutely at rest in the car or skip bucket.

A brief description of a new steam station of the Electric Bond and Share Co. might be of interest. The ash car is electrically operated, capacity $2\frac{1}{2}$ tons and the entire side opens so that the large clinkers will easily pass through. The skip bucket is three tons capacity so that it can take an entire carload.

The winding machine is spur geared type with gears enclosed and running in oil, and the entire machine is located in a clean space on the operating floor of the boiler room.

The control is Cutler-Hammer, and the skip is semi-automatic in operation, that is, the bucket performs a complete cycle starting from the pit ascending, dumping, reversing, descending and coming to rest in the pit. This control is placed on the winding machine itself and switches at the top and bottom of the runway for operating the skip are omitted. There is a safety switch at the top but it functions only as such. The ash bunker gates are all 48 by 60 in. and air operated.

George G. Bell.[1] Ash-handling troubles depend to a great extent upon the type of coal burned. Coal having a low percentage of ash which has a high fusing point will give comparatively little trouble; whereas coal with a high ash content, which fuses in the neighborhood of 2000 deg., cannot be handled with the same freedom from trouble; and the maintenance of the ash-handling equipment is certain to be very much higher than with the better class of coal.

The additional labor and maintenance resulting from burning coal containing a large percentage of ash having a low fusing point is caused by the formation of clinkers in the retorts and on the dump plate or clinker grinder pit as the case may be. Clinkers invariably result in an increase of the amount of coke in the ash. The practice mentioned by Mr. Hunter of admitting air to the ash-pit in order to reduce the amount of carbon in the ash is objectionable, as such burning out in the ashpit is apt to form a large clinker, which may bridge across the ash-hopper gate and require the use of slice bars to break it up before the ashes can be removed. If the ash is allowed to fill the pit till it comes in contact with the

[1] Mgr., Power Development, West Penn Power Co., Pittsburgh, Pa.

wind boxes and stoker dump grates, the maintenance on these parts will be exceptionally high.

The problem of properly quenching ashes is a difficult one. There is a great deal of trouble with nozzles; those giving the best results are probably of home-made construction, the holes being of liberal size. It is advisable to provide facilities for cleaning spray pipes with the boilers in service.

The combustible and corrosive gases, generated when the hot ashes are sprayed with water, will produce a pressure in the pit; and unless the ashpit gate has a water seal, these gases will be discharged into the ash cellar. With a sufficient water seal, the gases will be forced to escape through the dump grate into the combustion chamber, where the unconsumed gases will be mixed with excess oxygen and consumed. A water-sealed ashpit also prevents the clinkers formed over the nose of the stoker deflecting the partially or completely burned gases into the ashpit and out through the ash gate, which may in extreme cases make it impossible to work in the ash basement.

Regarding ashpit capacity, Par. 26 states that a continuous conveyor system will require a smaller ashpit. While this is permissible, an exterior storage of some type will probably have to be provided, in order to take care of contingencies which arise in the final disposal of the ashes. The advantage of a large ashpit storage is that it reduces to a great extent the liability of having to take a boiler off the line on account of the ashpit filling. This capacity should be large enough so that labor troubles will not result in closing the plant in a few hours.

Regarding the capacity of ash hoppers, Par. 27 should include an allowance for the coke in the ash. Some operators claim results as low as 10 or 15 per cent coke in the ash; but few persons would design an ash hopper figuring on less than 25 per cent.

The necessity of providing for the handling of large quantities of ashes together with the rapid deterioration of the iron work due to the corrosive action of the gases and the destruction by the ignited gases of the ashpit doors led to the adoption of the water-sealed type of ashpit originally proposed by Mr. Frederick Sargent, in the Springdale plant; and it is being installed in the 1922 extension to the Windsor plant. This later plant was originally designed with a maximum distance of 25 ft. between the top of rail in the ashpit and the boiler-room floor. A small-sized locomotive was used in this ashpit requiring a clearance of about 12 ft.

It was felt that a water-sealed gate was necessary; and when this problem was submitted to the gate manufacturers, and designs were submitted, the matter of the disposal of the water from the clinker grinder and the spray pipes in the ashpits was the main difficulty. The water conduits being small were very liable to clog up with small particles of ash. A comparison of the cost of this type of ashpit and gate with the type of water-sealed pit which could be installed led to the conclusion that the cost of the two schemes would be about the same, and that the water-sealed pit would have ten days' storage as against one day's storage with the other type. This cost included the cost of the cranes for loading out the ashes; but did not include the cost of equipment for removing the ashes from the basement to the outside disposal.

The Windsor arrangement differs from the Springdale arrangement in that a 3-yd. Blaw-Knox bucket is used instead of the $1\frac{1}{2}$-yd. bucket as at Springdale. The bucket moves lengthwise of the building only, so there is not the delay in dropping the bucket that there is at Springdale, as the motion at the latter plant is crosswise of the narrow pit. Also, the trough is continuous, so the water is all carried to a common point, the overflow being discharged into the sewer. The temperature of the water at Springdale remains at about 135 deg. with the boilers in operation. No air is handled through the ashpit basement. This is a matter which is very frequently overlooked in the ventilation of the building; and it is probable that if it was necessary to take air in through the ash basement, there would be less vapor formed from the water-sealed pit than from the ordinary ash hopper with water escaping from the gate. If the air supply for the boilers is drawn in through the ashpit basement, there is danger of these overflow pipes and even the gates freezing up when the boiler is banked for a long period.

For handling the ashes in the basement of the power house four cranes can be provided, if necessary; this means that each crane will have to handle a capacity of about 100 yd. per hr., or 50 tons. It is probable that if the capacity of this station is increased, the load factor on it will decrease, so it would look as if two cranes would handle the amount of ashes to be disposed of from this plant. However, twice this number can operate to advantage. This will permit of the removal of the ash in a single shift per day. The ashes are discharged by the buckets into a hopper and by means of rubber belts conveyed to an exterior ash

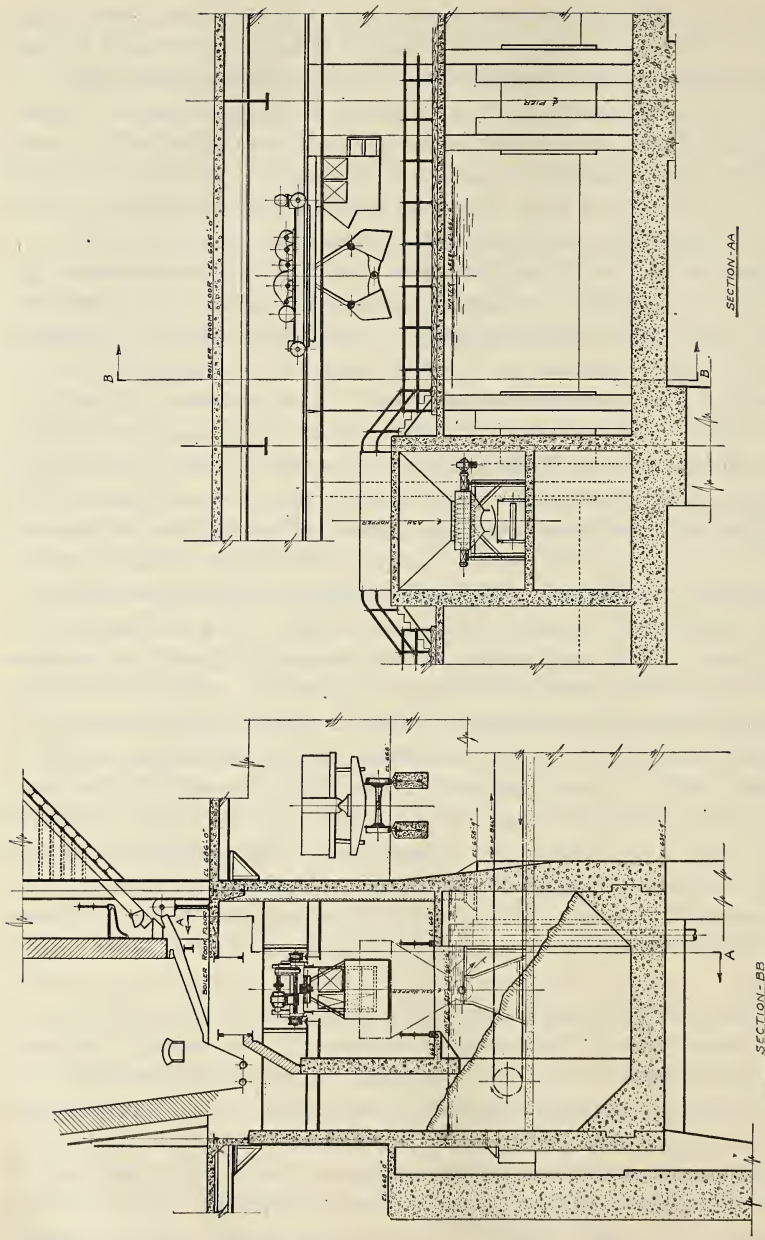


FIG. 40 WATER-SEALED ASHPIT, WINDSOR STATION

tank, from which they are loaded into 35-cu. ft. aerial tramway buckets. The capacity of the ash tank is 110 tons.

Arrangements are made that the ashes from the tank can be loaded into electric railway cars, steam railway cars, or trucks, as well as into an aerial tramway for ultimate disposal in filling deep ravines in the immediate vicinity of the power house. In case the belt conveyor is down, the cranes can discharge directly into railway cars located at either end of the ashpit.

With this system the main boiler-room floor is only 20 ft. above high water in the river; whereas if the ashes were to be dumped into small hoppers having not over a day's capacity, and be dumped from there into railway cars, the boiler-room floor would be located not less than 30 ft. above high water, and probably 35 ft. This reduction in the height of the boiler room can only be obtained without some additional expense in case the foundation for the boiler room is sufficiently below high-water level to permit of the installation of the water-sealed pits.

Mr. Funk stated that he did not know where to place the water-sealed ashpit with regard to reliability. I wish to point out that the same type of ash disposal equipment can be used as with the ordinary ash hopper, that is, railway cars, so this part of the system can be increased in units as the power house is extended; and that the investment in pits, including the cranes for handling, is no more than the expense of the modern ash-gate hopper. In a large power house sufficient cranes can be installed at a total cost not in excess of the ash hoppers and gates to allow for the breakdown of a portion of the equipment. The experience with cranes has been very satisfactory, and maintenance on them very low; whereas, our experience with large air-operated dump gates, handling partially quenched ashes, was very unsatisfactory. Maintenance on the railroad equipment is considerably less with thoroughly quenched ashes; and working conditions in the basement are so much improved that instead of ash-handling being difficult to keep men on, it is considered desirable work.

Eugene Hahn. In marine practice there are unavoidable losses of fresh water through safety valves, whistles, leaks, soot-blowers, and a number of incidental causes in the operation of a boat. The addition of any other apparatus that should require steam for its operation that could not be returned to the condenser would, therefore, appear to be unwise, unless there were good enough reasons

in its favor. The above unavoidable losses of steam in the operation of a boat are taken for granted, and there is usually a wide margin allowed for them, as the frequency of operation of those elements contributing to this loss of fresh water is highly uncertain. As there is no fixed relation between the horsepower rating of the vessel and the "make-up" water carried in the tanks, it is impossible to state whether the actual loss of fresh water contributable to the above causes is five or ten per cent of the full amount carried in the tanks. However, it is a fact that hardly ever has a vessel run out of "make-up" water due to the losses normally expected from above causes.

As against the uncertainty of the amount of loss of fresh water due to the above causes, the ejection of ashes by steam requires a predetermined amount, depending upon the quantity of coal to be consumed on a voyage, and its percentage of ash. This amount should either be so small that it could be well spared from the supply of "make-up" water carried in the tanks, or, should the supply be already scant, it is to be determined whether the advantages of a steam ash ejector are sufficient to warrant the fractional increase in the amount of fresh water carried in the tanks.

Every now and then we hear about a case where steam has been used instead of water with a hydraulic ash ejector. This usually occurs in case of breakdowns of ejector pumps that could not be repaired promptly during a voyage. However, upon arriving in port, this emergency method is soon discontinued, due to several obvious reasons. The nozzle that was good enough for water is far from the right size and proper design of a steam nozzle and will consequently use several times the amount of steam that it actually should, also with the ordinary hopper arrangements of a hydraulic ejector it is impossible to feed ashes and large clinkers at the greatly increased rate at which a steam jet can discharge the material as compared with a water jet. Also, with the customary design of a hydraulic ejector, it was impossible to utilize the combined advantage of continuous suction and of a steam jet, as against the pressure only of a water jet. Furthermore, the hydraulic ejector, having been able to resist the occasional "in-wash" of a rolling sea, could not operate with a steam jet under identical conditions, unless the height of the ejector was increased and its discharge end re-designed. Upon arriving in port the failure of such an emergency steam jet ejector was still more evident by the soiled condition of the paint, and even the deck of the vessel. This was due to the

fact that, there having been no water ejected with the ashes, the fine dust was permitted to settle on the outside of the boat. The elimination of the very reasons that discouraged the continued use of an emergency steam ash ejector naturally is appreciated by marine engineers, who look for something simple and more economical than the average hydraulic ejector. Such an improved steam ash ejector will, before all, reduce the loss of fresh water to a minimum. Its steam nozzle is designed on the best principles of turbine practice, and to reduce the time during which steam is passed through it, there is an automatic connection provided between the steam line and the starting and stopping of the ejector.

To prevent the scattering of fine ash over the boat, a connection is made between this automatic gear and water spray located at the discharge end of the ejector, and supplied with sea water. In order to prevent the "in-wash" of a rolling sea the discharge end of the ejector has been raised well over the highest water line, and a self-sealing sea-gate has been provided, which is also connected with the automatic gear. This inter-connection of different elements that have proven so objectionable on emergency steam ejectors leaves nothing to the fireman or stoker to forget, — this being so very important in marine practice.

On an actual test with a flow meter it was found that a ton of ash with clinkers up to 5 in. in size has been discharged in eight minutes at a total expenditure of 120 lb. of steam, at 200 lb. pressure. The minimum diameter at its throat was $\frac{11}{32}$ in., and the height of the ejector itself was 22 ft. Based on these figures it is easy to estimate on the total requirements in fresh water which cannot be returned to the condenser during an average voyage. Should an 8000-ton boat consume about 500 tons of coal on a given voyage, producing, at 12 per cent, 60 tons of ash, the total amount of fresh water required and lost would amount to 7200 lb., which is about $3\frac{1}{2}$ tons. The average vessel of this size will carry about 300 tons or more of fresh water in its tanks, so that the total and definite sacrifice in fresh water will be somewhat over one per cent of the water carried in the tanks. The coal required to evaporate 7200 lb. of water at an evaporation of 8 lb. will be about 900 lb., or approximately half a ton.

The greatest single return for this apparent sacrifice will be the saving of that amount of coal that must be saved by not requiring the ejector pump to operate. The amount of steam required by this pump is usually problematic, and depends upon its size, its

distance from the ejector proper, and its condition of repair. Other elements entering into the steam consumption of this pump are the frequent clogging of the ejector pipe, — during which time the pump often continues to operate unnecessarily — and other possible disorders with both the pump and the ejector. Due to these variables, and the fact that no reliable figures are available as to the actual steam consumption, it is assumed that about two to five per cent of the total coal consumed on a voyage could be saved by replacing the hydraulic ejector with a good steam jet ejector. On this basis in above example, about 10 to 25 tons of coal could be saved out of the total of 500 tons consumed.

Other advantages in favor of a good steam jet ejector are:

Extreme simplicity of construction, — so very important on board ship,
Considerable reduction in cost of repair and maintenance,
Elimination of a cumbersome ash hopper in the crowded stoke-hole,
Prevention of flooding of stoke-hole,
Considerable reduction in time and labor required for ash ejection.

It is evident from the foregoing that the application of a steam jet arrangement for ash ejection on board ship is both feasible and advantageous, provided the proper requirements of marine practice are fully met.

The writer wishes to confine his remarks to what the authors refer to as "Air Conveyors" only. While it was admitted that these conveyors have their great merits from a number of view points, it seems that the general impression gained by most of us is that they are anything but economical in their steam consumption. To best offset this assumption, the writer wishes to refer to a number of installations where a considerable reduction of steam consumption was achieved solely by re-arranging the layout of the conveyors. Unfortunately, however, this important matter has been given little consideration in the past, so that there is little wonder that steam jet conveyors became known as a wasteful apparatus in steam.

Steam jet conveyors, — contrary to the so-called vacuum conveyors, — if properly laid out will fully utilize the thermodynamic action of the steam jet, both in the suction and pressure line of the conveyor. Steam leaving the exhaust nozzle of a so-

called vacuum conveyor will always be of higher pressure than it should be in case of a steam jet conveyor, which, of course, proves the respective efficiency of either type.

Years of experience in this particular field have proven to the writer that the average engineer is not expected to know the intricacies of ash handling by the steam jet principle, and that it requires an expert in this line to pass on suggested layouts. To elaborate on this subject would be beyond the limit of this discussion at this time.

In the case of the marine ash ejectors, it was shown that the proper design of the steam nozzles is of extreme importance as a means of reducing considerably the steam required. This applies to stationary practice also. Assuming that a conveyor has been laid out to best advantage, and the steam nozzles have been designed properly, the only important requirement left is the proper design and material of the different fittings. These should be so designed that they will be easily accessible in case of repairs or replacement of their different wear liners and be cast of hard enough iron to resist abrasion successfully.

It remains to those interested in the greater popularity of steam jet ash conveyors to point to the increasing number of satisfactory installations both from the standpoint of low steam consumption, as well as proper design of the layout and different fittings. The growing interest in this type of ash conveyors proves that they are on the right track.

Charles E. Prout. Regarding Par. 15 of the paper, the manufacturers of G & G telescopic hoists have developed methods of handling ashes for the types of buildings for which the authors state no satisfactory means of handling have been found.

The ashes are usually placed in cans and delivered at the point where they are to be raised to grade level; and, as stated, the problem is really that of hoisting the ashes from the boiler room to the street. The block and tackle is out of the question because it is slow and the factor of labor too high.

With this hoist and its supplementary equipment, comprising a complete unit, it is possible for one or two men to take care of the entire work of removing ashes with only a small labor cost. There are four principal models, one of which will meet the particular conditions in such buildings as hotels, schools, hospitals, office buildings and factories, where the ash removal problem is as previously stated. There are two general divisions of conditions.

First: In a hoistway so located that a wagon or railroad car can drive alongside. The depth of the area and the distance of lift are the factors that determine whether electric or hand power should be used.

The normal position of this hoist when not in use is underneath a pair of checkered steel or vault light sidewalk doors about 4 by 4 ft. square. The first step when it is desired to remove ashes is for the operator in the hoistway to turn a crank which raises the hoisting head into working position. This operation automatically includes the gradual opening of the steel doors over the hoistway and when the hoist has reached its extended position the doors are automatically locked open. Gates permanently attached to the leaves of the door, guard the hoistway, preventing the possibility of passersby falling into the pit. One of these gates is a fixed member and the other (having spring hinges) can be opened outward, but cannot be swung inward.

The overhead crane is raised and held in position by a strut and the hoist is ready for work. With this equipment it is customary to use only two men. One man can operate the motor and take care of hooking full cans and unhooking empty cans as they are raised and lowered. The average can filled with ashes weighs about 150 lb., or approximately 3 cu. ft., and can be raised and lowered at a speed of not less than 60 ft. per min. With the use of swing bail cans supplied with the hoist, the man above can empty the ashes directly into the truck without unhooking. The hoisting head swings on ball bearings from a point directly over the hoistway to a convenient position over the truck, so that practically no effort is required to swing the can around. When the ashes have all been removed the overhead crane is lowered and the entire apparatus telescopes below the grade level. Sidewalk doors are automatically closed and locked. The spring guard gates fold below the grade level. The amount of current consumed by this apparatus is surprisingly small, as shown by various tests.

Second: In buildings where conditions are similar, except that it is not possible for a truck to drive close up to the hoistway, a telescopic electric hoist, which has not the overhead crane feature, may be used. The motor in this case is located on the hoisting head and controlled at the grade level. Very recent improvements in this type of hoist include a gravity lowering device so that ash cans (or any load up to 500 lb.) can be raised at a speed of 60 ft. per min., and lowered by gravity under brake control. This gravity

lowering feature saves current and increases the speed in handling large quantities of ashes.

Regular monthly lubrication is really all that is required. This ash-removal equipment is particularly desirable for use in schools, hospitals, and other places where any unusual amount of noise must be eliminated.

For smaller plants these two types of hoists (cellar to truck height and cellar to grade level) may be obtained for hand power operation; but for buildings where the real problem is that of hoisting large quantities from the boiler room to the street, the electrically operated types which have been developed after many years of study and are especially made for handling ashes, are most satisfactory.

While the apparatus used by the Union Electric Light and Power Company, St. Louis, Mo., as described in Par. 16, might have been an improvement over former methods used, the operation described could not have been speedy. For the average plant operator it has been our experience that the service rendered by such equipment would be entirely too slow and otherwise unsatisfactory.

The Authors are gratified by the very interesting discussion which shows the great interest taken in the subject.

As Mr. Marsh has said, the 8-in. air conveyor is about the maximum size for economy of operation. Above this size, the capacity is greater than that of the man feeding it; and as the steam consumption is directly related to the capacity, there is considerable waste of steam. The highest economy is attained when the capacity is kept down to that of the operator. Similarly, shutting off the steam by means of local controls when ashes are not actually being fed results in enormous reduction in steam consumption.

Mr. Allen's discussion of the difficulties of quenching ashes and of his improvements in sprayers is very welcome. The "efficiency" of ash quenching is very low, as it is impossible to get the water exactly where it is wanted except when the ashes are actually drowned out as in Figs. 16 and 22. Any method which results in getting the water to the hot ashes only will save a large proportion of the cooling water.

While Mr. Funk generally confirms the authors' conclusions, it seems advisable to refer to Par. 90 describing the conveyor illustrated in Fig. 22 where it is pointed out that in some cases "only

a single conveyor is used with auxiliary chutes for use in case of breakdown." Ashpit storage is not practiced with these conveyors. With some forms of continuous conveyor, ashpit storage is desirable in case of breakdown. It is a matter of judicious correlating of the different factors.

Trouble with gas explosions in the storage tanks of air conveyors was experienced with some of the early installations, but this difficulty appears to have been entirely surmounted. When wear occurs in places where there are no wearing backs, it is obvious that they should have been provided, and this is one of the points which are being improved in the development of the air conveyor.

Replying to Mr. Elmer, some powdered coal installations result in the ash being melted to a slag which can be tapped off as in a blast furnace or foundry cupola, or allowed to cool and be broken out with the expenditure of much labor. The use of the water screen at Milwaukee results in causing the ash to be deposited in a fine powder which is easily handled with an air conveyor. Reference should also be made to the reply to Mr. Beaumont.

Mr. Moultrop well summarizes the salient problems of the modern power station as those of water, coal and ashes; and points out that the final disposal is one of the most serious features of the ash problem. But it is not convenient to discuss ash disposal in a paper of this kind, because that is a matter which is governed by the environment of the plant and which must usually be determined as one of the primary conditions of operation. The mechanical features of handling the ashes must then be decided upon to suit this condition, and they may even have to be sufficiently flexible to meet a change to a different condition of disposal at some future time. The development of mechanical contrivances for handling ashes, may, however, have a distinct bearing in widening the field for final disposal as in the case of the Springfield and Windsor plants of the West Penn Power Co.

The remarks of Mr. Maynz confirms the authors' views on the quality of experience and judgment which should be brought to bear in deciding on an ash-handling system. Headroom is expensive, and the usual engineering compromise must be made between first cost and cost of operation; but "in designing new plants ample headroom should always be provided," as stated in Par. 32.

Mr. Libby's experience also confirms these views and shows the necessity of full investigation of the problem before proceeding with the design of the apparatus.

Mr. Tenney clearly shows how each individual plant must be separately considered and that method adopted which will most suitably meet the particular requirements. He also shows how these requirements have been met by the adoption of different methods in different plants.

The freezing of the ash bunker gates in winter and the method of thawing them is of particular interest. The wet ash in the hopper may also freeze in some cases, and have to be thawed out. Climatic conditions must not be neglected in ash handling as these may have considerable bearing on the methods available.

Mr. Rankin Eastin's installation is a very successful example of water sluicing, and is the more creditable as he originated the scheme without knowing of any similar system elsewhere and therefore without the guidance of the experience of others.

Mr. Prout's description of the telescopic ash hoist which has been developed for hotels, office buildings and other cases where ashes have to be lifted from basement boiler rooms to the street level is of considerable interest and may be compared with the earlier attempts such as that described in Par. 16. The low cost of operation is worthy of attention.

Mr. Beaumont's description of an ash car and skip hoist installation is a valuable addition to the discussion. But in championing the skip hoist, Mr. Beaumont should not feel disappointed that the paper was not more completely devoted to it as the object was to describe the various systems in use and not one particular method. The authors are in accord with him as to the simplicity and flexibility of the ash car and skip-hoist as well as to its capacity for extension, as is obvious from the paper.

As to the size or absence of a hopper with continuous conveyors, the reply to Mr. Funk applies equally here.

That steel-plate hoppers have been longer in service than those of cast-iron plates does not mean that experience with these materials is limited to the relative duration of their use in this manner. The materials have been used in comparison in situations where the conditions are quite similar to those of this service, and the greater durability of cast iron over wrought iron and steel is very well known indeed. Of course, it is obvious that with adequate protection, the steel plates will not be corroded; but experience with protective coats generally makes it wise to choose materials which will suffer the least if the protection fails.

The passage of large clinkers through the ash-handling system

to final disposal is not always either convenient or desirable. In some instances a conveying system which will not handle large clinkers is preferable owing to its other advantages, and large clinkers must either be broken by hand or by mechanical crushers. Also, there are cases where ashes can be utilized if small, but not when containing large clinkers, and this may interfere with ash disposal.

The relative movement between the ashes and the conduit in air conveyors has been amply considered in the paper. The advantage of having no moving parts lies largely in the greater amount of wear which stationary parts will usually stand before needing renewal and in the ease with which erection and renewals are handled. Moving parts require much more mechanical skill and time for their proper fitting to enable them to work properly without undue friction and breakage, than stationary parts.

With reference to the Lakeside plant of the Milwaukee Electric Railway and Light Co., Mr. Beaumont has inadvertently appeared to discredit modern powdered coal burning through lack of information. The skip hoist he mentions has nothing to do with the subject of ash handling. As Mr. John Anderson has just written us, "it was put in to act as an elevator to the basement for handling material that an air conveyor system was never designed to handle. This includes castings, brick for furnace repairs, and any other material that is used in the basement of a boiler room for the maintenance and operation of the equipment in such basement." As is stated in the paper, all the ash at Lakeside is handled by air conveyors.

Mr. Bell's comments are very valuable and generally accentuate the authors' views as expressed in the paper. His detailed description and comments on the Springdale and Windsor plants should be very highly appreciated, and reference to Fig. 16 may well be made in studying his discussion.

One or two of Mr. Bell's remarks call for comment as being evidently based on misunderstanding. From Par. 38, it will be seen that the authors condemn the admission of air to the ashpit which results in combustion therein, just as strongly as he does. Also, reference to Par. 27, will show that the authors wrote that in addition to other things "allowance must be made for unburned coal coming in with the ash." This allowance should not be less than 25 per cent, as Mr. Bell says.

Mr. Eugene Hahn makes a strong plea for the use of the air

conveyor on shipboard. As he well says, the disadvantage of the loss of water must be weighed against the obvious advantages. But the air conveyor in good condition should not be compared with the water ejector with its pump out of repair. The two kinds of apparatus should be compared when in a similar and reasonable condition. As pointed out in the paper, the very simplicity of the air conveyor sometimes leads to its neglect.

While the test mentioned showed only 120 pounds of steam per ton of ash, it is generally found in land practice that the steam consumption under regular working conditions averages considerably more than this. The analysis of a definite case, however, would not be invalidated if based on the usual figures obtained from stationary plants.

The discussion of marine practice was not undertaken at any great length in the paper. The subject is one well worthy of more deep study and Mr. Hahn's contentions should not be lightly dismissed.

Mr. Hahn's remarks on the thermodynamic action of the steam-jet in the two types of air conveyor in general use are open to contention. The steam may be expanded to a lower pressure in a vacuum conveyor than in the so-called steam-jet conveyor, but whether this is done in actual practice depends upon the design of the nozzle in each case and not upon the conveyor system.

The authors wish to express their full appreciation of the great trouble taken by those who have contributed to the discussion and so worthily amplified what was written in the original paper.

No. 1867*a*

THE DEVELOPMENT AND USE OF THE MODERN CHAIN GRATE

By T. A. Marsh, East Chicago, Ind.
Member of the Society

After discussing the need of progressive combustion of the several constituents in coal, and the variations in grate design to handle different coals, the author takes up the improvement of chain-grate practice as regards air-tightness, cooling, size of combustion space, etc. He shows how greater boiler capacity can be obtained by increasing grate size or by more intense natural or forced draft (0.50 to 0.60 in. for a combustion rate of 40 to 45 lb. per sq. ft. of grate per hour), but states that high and sudden overloads are best met by forced-draft chain grates burning 55 to 60 lb. bituminous coal per sq. ft. per hour, with air at 2 in. pressure distributed through compartments between the runs of chain. He describes such equipment, and states that 200 per cent of rating can be reached in 8 min., from a short-banked fire, or in 52 min. from a cold grate. The paper concludes with a table of results obtained in a number of stations equipped with chain grates.

ANY STUDY of the adaptability of various stoker types and their proper functioning as fuel burners must start with an analysis of four constituents in coal, namely, moisture, volatile, fixed carbon, and ash; or roughly, water, tar, coke and dirt. These constituents are so dissimilar that each must be differently treated in the combustion process, and specific provisions for burning them in proper sequence must be provided in stoker and furnace design. These facts at once determine as fundamentally correct those types of stokers embodying the principles of progressive combustion, by which is meant in practice a continuous movement of coal through the furnace and the providing of definite treatment at proper time and place for the burning of each constituent of the fuel.

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Discussion on paper No. 1867 *a*, *b* and *c* follows paper No. 1867 *c*.

2 Chain grates embody the principles of progressive combustion. As coal is fed into the furnace, the moisture is vaporized; then the volatiles are distilled and burned in suspension, followed by the burning of fixed carbon. Finally the ash is discharged over the rear, and the air space of the grate is automatically cleaned for a repetition of the process.

3 Variations in the proportions of these constituents are seen in coal from different sources. Some coals are more free-burning, others tend toward coking. Some have high percentages of ash, others less. Differences as to clinkering or non-clinkering are noted. Such variations at once emphasize the fact that no one stoker and furnace can be suitable for all coals. Stokers and furnaces adapted for progressively burning certain coals must be modified if they are to be used for burning other coals having widely different proportions of characteristic constituents.

4 Free-burning coals burn best when the fuel bed is undisturbed; coking coals demand fuel-bed agitation. High-ash coals demand continuous scavenging of refuse from the grate; with low-ash coals this is not necessary. Clinkering coals must not be agitated, or clinker formations will result. With non-clinkering coal, agitation has a less detrimental effect.

5 Chain grates were first developed to burn high-volatile, free-burning, high-ash, clinkering coals. For such coals they offer in best form the specific combustion treatment demanded by the characteristics of such coals. The fuel bed is undisturbed, making this stoker type suitable for free-burning and clinkering fuels. The ash is continuously discharged from the grates, which is necessary for continuous operation with high-ash coals.

6 The development resulting in the modern chain grate has been gradual but definite, and has involved the successful burning of many coals having far different characteristics. All successes, however, have been dependent upon the application of the principles of progressive combustion. Coal burning in any practical process involves some losses: a total conversion of the heat in coal into heat in steam is impossible. Costs of operation must also be considered. These facts determine that every combustion process must finally operate with a common-sense balance between the three important factors in steam generation economy: namely, maintenance, efficiency, and capacity.

MAINTENANCE

7 Maintenance and reliability are closely related. The chain grate, with two-thirds of the grate surface out of the furnace and with a simple driving mechanism, was always low in maintenance costs. In the earlier design, however, some parts were exposed both to heat and wear. Parts necessarily exposed to wear or strain often did not have sufficient resistance.

8 The modern chain-grate designer recognizes that the functions of stoker parts fall into three classes: Supporting the fuel, actuating the mechanism, or furnishing the necessary support for coördinating the first two. Parts performing any one of these functions must be protected as far as possible from failure due to being called upon to perform any of the other functions. Actuating mechanism is accordingly removed from heat, and is subject only to wear. The chain, which is the fuel-carrying part, is now subjected to but slight heating as a result of adequate ventilation. Stoker frames are not subject to wear and are not exposed to heat. When exposed to heat or wear, replaceable parts are provided.

EFFICIENCY

9 The high fuel costs of recent years, combined with a broader knowledge of combustion principles, have led to the development of features for improving efficiency. Inasmuch as chain-grate stokers have always involved very low auxiliary power consumption — about 0.5 hp. per stoker — all results obtained are practically net, and little improvement could be hoped for in that direction. Efforts were therefore directed toward the elimination of other losses, such as excess air, and to the reduction of furnace and ashpit losses.

IMPROVEMENTS IN GRATE DESIGN

10 *Air Leakage.* A grate surface moving through a furnace presents the problem of air seals along the sides of stoker and at the rear. The reduction of air leakages at these places has contributed materially to modern economy.

11 Overhanging bridge walls and water backs are necessary. All attempts to approach efficient combustion without water backs have failed. Many designs of water backs were tried, resulting

finally in the modern water back connected into the pressure circulation of the boiler. Water backs are successful only to the extent that they close the openings and eliminate air leakage between the bridge wall and the fuel bed.

12 Ledge plates or seals along the sides of the stoker went through a similar development, resulting in the modern adjustable ledge plates for making a proper rubbing seal with the stoker chain, and including means for adjustment to keep this seal intact.

13 The development of the ledge-plate flange adjustable against the side of the upper chain made it possible to raise and lower the stoker to vary the discharge opening under the water back in order to meet broad changes in fuel conditions. The improved results thus obtained indicated the desirability of a water back adjustable as to height, and several designs were developed. These were for the most part difficult to construct or to maintain. From such designs the modern fuel retarder was developed. This is an adjustable member in connection with a fixed water back and bridge wall. The fuel retarder can be raised or lowered to make a definite air seal with the fuel bed.

14 Sifting of coal through the grate was an objectionable feature of early chain grates. This often amounted to from 5 to 10 per cent of the coal fired. The most important item in reducing this sifting has been the use of longitudinal skids to carry the chain in place of the cross-rollers employed in the earlier designs. Skids reduced the siftings to 1 or 2 per cent, and in addition to labor saving a more uniform fuel bed results.

FURNACE DESIGN

15 Simultaneous with the development of the stoker has been that of furnace design. Furnace volumes have been quadrupled during the last 10 years. Modern chain-grate furnaces have 2 cu. ft. of furnace per developed horse power. Losses due to unburned hydrocarbons have been greatly reduced.

16 Early chain-grate furnaces had arches of limited height and length, and with insufficient draft could burn fuels at rates which modern practice would consider as low. A rating of 150 per cent was exceptional. Modern high-set arches with ample draft and adequate furnaces permit high combustion rates and high ratings from boilers.

17 Improved furnaces, proper draft, longer arches, water backs, and fuel retarders have all contributed to the reduction of ashpit losses. High stoker and furnace efficiency is dependent, not on a minimum of any one loss, but on a minimum sum of the losses due to excess air, ashpit loss, and unburned hydrocarbons. The modern chain grate and furnace permit adjustment and continuous operation, with the aggregate of these losses a minimum.

18 Chain grates are ideally adapted for producing smokeless combustion, due to the uniform feed, uniform fuel bed, and the arch. With modern furnaces and proper air control, smokelessness within the capacity of the furnace can be assured even with the high-volatile smoky coals of the Middle West.

CAPACITY

19 Modern steam turbines called for greater capacities from boilers. This demand was met in the chain-grate field by development along three general lines:

a Higher drafts
b Larger grate surfaces
c Forced draft.

20 *Higher Drafts.* Chimney heights for natural draft were increased to 200, 250, and even in excess of 300 ft. Many plants installed induced-draft fans. Furnace drafts of 0.50 to 0.60 in. were obtained, and combustion rates reached figures of 40 to 45 lb. per sq. ft of grate surface per hour.

21 *Larger Grates.* This rate of combustion seems to be about the limit of natural-draft performance. Efforts to obtain higher ratings have led to larger stokers. Stokers were built up to 18 ft. long, with ratios of grate to heating surface increased to 1 to 30. With such grates, and high draft, boiler ratings up to 200 per cent were possible.

22 *Forced Draft.* The need of still higher ratings, particularly with low-grade coal, brought the application of forced draft to the chain grate. Early installations applied forced draft at uniform pressure under the entire grate surface. This increased capacity, but did not control excess air. Forced draft under a fuel bed with the slightest tendency toward thin spots caused excess-air losses, chilled the furnace, and reduced capacity.

FORCED DRAFT WITH COMPARTMENTS

23 The logical development for air control was to divide the stoker into compartments and control the air supply to each compartment.

24 In the early forced-draft chain grates difficulty was experienced in obtaining uniform air distribution laterally. This was obviated by some designers by reducing the air space in the grate surface to a very small amount and depending on the resulting chain resistance to equalize the pressure under the grate surface. This produced the desired results, but the required higher pressure in the air compartment entailed higher auxiliary power for fans. Small air spaces greatly reduce boiler ratings when operating on natural draft.

25 Other manufacturers solved the problem of obtaining uniform air distribution by the use of more air space, more liberal air ducts, lower velocity, and lower air pressures.

26 The forced-blast chain-grate stoker has been in commercial use for some twenty-five years, and in successful use for approximately ten or twelve years in the anthracite-coal regions, but it was not until five or six years ago that any really successful installations were made for burning bituminous coal.

27 Several chain-grate manufacturers had applied forced blast to their natural-draft stokers, and had increased the amount of coal burned per square foot of grate by so doing, but it was not until uniform air distribution was obtained and blast compartments between the runs of chain had been arranged for the control of the blast, that the performance of this type of stoker began to attract the attention of the bituminous-coal user.

28 The general design of the forced-blast chain-grate stoker is quite similar to that of the natural-draft chain grate with which every one is more or less familiar.

29 This stoker consists of a continuous or endless chain of links or grate bars, traveling over front and rear sprockets. Side girders are used in some cases, but in others are entirely omitted and transverse members forming compartment sides and wall supports substituted therefor. Above the front sprockets is a sizeable coal hopper, and at the back of this hopper a stoker gate which can be raised or lowered to vary the thickness of the fuel bed on the grate.

30 The stoker is usually set in an extension or Dutch-oven furnace, the arch of which radiates its heat on the incoming fuel under the stoker gate and causes it to ignite, the fire gradually burning down through the fuel bed until it reaches the grate. The thickness of the fuel bed is governed by the character of the coal burned, the length of the stoker grate, and the capacity at which the stoker is to be operated. With the same grate, and with different grades of coal or different capacities, the fuel bed may vary between 5 in. and 9 in. in thickness.

SIZE OF FORCED-DRAFT STOKERS

31 The length of the stoker, which should be taken as the distance between the inner face of the stoker gate and the sealing plate of the rear blast compartment, is varied somewhat to conform with the type of boiler under which the stoker is installed, and with the rate of combustion which is desired. Very few forced-draft chain grates are less than 12 ft. in length, but only in exceptional cases do they exceed 20 ft.

32 The width of the stoker is usually made slightly less than the boiler-furnace width, so as to afford column protection. Single stokers practically the full width of the furnace have been built, even for the wide central power-station boilers now so popular. There are forced-blast chain-grate stokers in operation today that are 24 ft. in width, and widths of 27 ft. to 30 ft. are under consideration.

33 Various types of links are used with the forced-blast chain-grate stoker, some of which are held in place with steel link rods, while others are strung on malleable-iron dovetails, which in turn are bolted to carrier bars extending the full width of the stoker, and attached to forged conveyor chains. Regardless of the type used, however, the design of the link is such that relatively fine jets of air are admitted through the grate, and a uniform air distribution is applied to the fuel bed from any given blast compartment.

34 The forced blast is usually admitted into five or six compartments formed between the runs of the chains. These compartments are sealed from one another, and dampers or slides are arranged so that the blast to any one compartment may be regulated independently of that in any other compartment. Some

designs are so constructed that any compartment can be operated on forced or natural draft, or closed off entirely, and so that the change from one to either of the other conditions of air supply can be made simply by the operation of a single adjustment lever.

35 As the coal is burned, the resistance of the fuel bed decreases toward the rear end of the grate, and the air pressure is varied in the different compartments to meet the requirements and effect proper combustion at any given point, with least excess air. It is possible to use only half or three-quarters of the grate length when operating the stoker at low capacities, and still maintain a comparatively high rate of combustion on the front portion of the stoker.

36 The blast pressure in any one compartment rarely exceeds 2 in. water pressure, even at a combustion rate of 55 to 60 lb. of coal per sq. ft. of grate surface. Low-pressure designs obtain similar combustion rates with about one-third of this pressure. The highest pressure is usually carried in the second and third compartments from the fronts of the stoker, as it is at these points that the fuel bed on the grate is completely ignited and the maximum rate of combustion takes place.

37 The blast to each compartment is regulated by hand control. The dampers or slides to these compartments, however, may be set in a definite relation to each other, after once determining the requirements of blast for a given fuel; and the rate of combustion is regulated by varying the speed of the fan or by operating a single damper at the forced-blast-fan discharge, or in the main air duct, and by adjusting the boiler damper and fuel fed to correspond. The amount of blast used under the grate and the speed at which the stoker chain travels usually determine the capacity at which the stoker may be operated with a given thickness of fuel bed.

38 The small amount of fine coal and ash siftings that falls through the upper run of the grate into the blast compartments is handled in different ways by different manufacturers. With some, the siftings are raked from the blast compartments through doors on the boiler-room floor; others arrange steam jets to blow these siftings to the end of the blast compartments, where they are removed by steam ejectors. Still others construct the stokers in such a manner that the siftings fall on to the lower or return run of the grate, and are carried forward to the front end of the stoker,

where they are deposited into a hopper, from which they are conveyed to the coal bunker or ash pit.

COMBUSTION ARCHES

39 Stokers installed for the burning of the Central States high-volatile coals have ignition or combustion arches set about four feet above the grate at the front end, and somewhat higher at the inner end. These arches cover approximately 50 per cent of the length of the stoker, are of the suspended type, and are made from high-grade refractory material.

40 Stoker installations for the burning of the low-volatile high-carbon bituminous coals of the Eastern States use a shorter arch, covering about 30 per cent of the stoker length, but set at practically the same height above the grate as for the high-volatile coals. Arches set at this distance above the grate require a short ignition arch, 12 to 18 in. in length, at the stoker gate, and set 18 to 24 in. above the grate. The increased rate of combustion obtained with forced-blast chain grates necessitates a better grade of refractory for the entire furnace lining.

41 Various schemes are being considered today for cooling the furnace side walls of all the different types of stokers operating at high rates of combustion, with the purpose of reducing furnace maintenance costs. Of these various schemes, those employing water-cooled members will undoubtedly prove the most successful, because of positive functioning.

42 Prevention of the formation of clinkers at the fire line on the side walls is a problem separate and distinct from reducing side-wall maintenance. Some manufacturers are using specially designed firebrick blocks at the fire line, through which blast from the blast duct enters the furnace. The air passing through these blocks keeps them cool and prevents clinker from adhering to them. Other manufacturers are using side-wall water boxes at this point, some of which are connected into the boiler circulation, while others are independently connected.

43 The bridge-wall water back is a part of the standard stoker equipment with several of the forced-blast chain-grate stokers. This water back is carried transversely across the rear end of the stoker 3 to 7 in. above the grate, and acts not only as an air seal at this point, but protects the bare links of the stoker from the reflected heat of the bridge-wall brickwork.

RESULTS SECURED

44 The forced-blast chain-grate stoker is particularly adapted to the use of free-burning coals that require agitation of the fuel bed to break the crusting or caking action so often encountered with the slower-burning coking coals.

45 With free-burning coals, which usually run high in sulphur, and have an ash with a low fusing point, results are being obtained, with uninterrupted operation, that equal the results obtained with other types of stokers using the low-volatile high-carbon coals.

46 When burning No. 3 buckwheat or coke breeze, fires 3 to 6 in. in thickness are carried on the grate and sufficient blast is carried in the first and second blast compartments to cause a gentle boiling or dancing of the fuel bed. Care must be taken, however, to regulate the thickness of fire and the blast so that excess carbon monoxide is not produced and continued combustion carried through the boiler to the chimney.

47 The forced-blast chain grate may be operated at continuous high overload capacities for days at a time with only slight variations in the steam output of the boiler. This operation is made possible by the uniform thickness of the fuel bed, the uniform blast pressures carried, and the uninterrupted disposal of ash and refuse from the grate. Such stokers may be operated efficiently under natural draft when burning as little as 10 lb. and as high as 35 lb. of coal per sq. ft. of grate per hour. Forced-draft combustion ratings reach 55 to 60 lb. of coal per sq. ft. of grate per hour.

48 The forced-blast chain grate is simple to operate and flexible in handling varying load conditions. If properly handled it can be brought from banked fires to full load in time to meet any ordinary power-plant requirements. A cold boiler can be put on the line in 45 min. from the time of lighting fires. The boiler can be operating at 150 per cent of rating in 5 min. from the time it goes on the line, or it can be operating at 200 per cent of rating in 7 min.

49 With the ordinary short-banked fire, which consists of a bed of coal three to four feet in from the stoker gate, the remainder of the grate being bare, the boiler can be brought up to 200 per cent rating within 25 min., while with the long bank, in which 50 per cent of the grate is covered with fire, the boiler can be brought up to 200 per cent rating within 6 to 8 min.

50 Automatic control has been successfully applied to both natural- and forced-draft chain grates. With natural-draft chain grates the usual control is by means of a steam-pressure or steam-flow regulator controlling the boiler damper and the speed of the stoker engine or motor. With forced-draft chain grates several control systems are in successful service. The electrical control is applicable on motor-driven installations. The steam-flow regulator controlling the air supply to the stoker and the speed of the grate is giving successful results.

51 The steam-pressure regulator controlling the air supply to the stoker and speed of the grate is also giving excellent service. In case constant furnace draft is desired, the balanced control can also be applied, as is frequently done with the two latter-mentioned systems.

DESIRED FEATURES

52 Among the features that it is desirable to embody in modern forced-draft chain grates and furnaces are included the following items:

Reliability. A rugged, heavy stoker with simple parts protected from heat effects and accessible for inspection and repairs

Maintenance. Heavy construction; all parts accessible; small parts exposed to heat easily replaceable; side walls protected by water-cooled members connected into the boiler circulation

Efficiency. Continuous fuel feed and ash discharge. Each part of the combustion process carried out in a definite manner, and a place designed therefor. Excess air excluded by definite seals in contact with the fuel bed. Ashpit losses low

Capable of control and adjustment, and of continuation in service under such adjustments. Low banking loss

Low labor of operating due to continuous process

Low air pressures so that auxiliary power may be at a minimum when forced draft is used

All compartments tight initially, and capable of being maintained tight, due to design and accessibility of dampers

TABLE 1 FUELS USED AND RESULTS OBTAINED IN ELEVEN BOILER PLANTS EQUIPPED WITH CHAIN GRATES

Description of Station	Location	Name of Coal Used	Moisture, per cent	Ash, per cent	Volatile, per cent	Fixed Carbon, per cent	Sulphur, per cent	B.t.u. per lb., Commercial Basis	Average Rating, per cent	Maximum Rating, per cent	Banked Boiler Hours, per cent	Economizers	Combined Efficiency of steam-generating-unit, per cent	Time covered
Central station [1]	Middle West	Illinois	14.5	16			4	10000	225	275	8	Yes	73.6	3 months
Central station	Middle West	Illinois	14.5	16			4	10000			30	50 %	69.4	3 months
Central station	Southwest	Kansas	4.66	11.7	33.05	50.62	4.2	10561				No	69.4	24 months
Central station	West	Iowa	16.51	18.72				8891	180		32	No	73	12 months
Central station	Middle West	Indiana	9.8	17.7	33.3	49	4.2	10000	170	230	30	No	70	12 months
Steel plant	Middle West	Indiana	18	12.2	35.2	52.6	2.20	10150	140			No	71	12 months
Steel plant	Pittsburgh district	Pennsylvania	2.09	10.5	33.7	55.8	1.60	13250	107			No	69.4	12 months
Central station	West	Colorado	23	12.5	32.5	32	0.20	8865				Yes	72	1 month
Central station	West	Kansas	6	19	27	48		10700	130	200	49	50 %	69	1 month
Central station	Middle West	Ohio&Penn.						11766	145	225	6	Yes	73.8	5 months
Central station [2]	Middle West	Illinois	14.5	16			4	10000	271			Yes	78.7	1 month

[1] Part of the stokers in this plant are forced-draft units.
[2] All of the stokers in this plant are forced-draft units.

The fuel bed surrounded on its active sides by water-cooled members.

Capacity. Capable of reaching high ratings by increasing combustion rates rather than by increasing effective grate surface in service. A stoker capable of operating on natural draft at ratings of 150 to 175 per cent and at higher capacities when forced draft is applied.

NATURAL DRAFT VS. FORCED DRAFT

53 The above specifications for an ideal forced-draft chain-grate stoker are such as produce best economy in steam production.

54 Whether natural- or forced-draft chain grates should be installed depends upon conditions in the individual plant. The station load, banking periods, boiler absorption with or without economizers, and the limits of the draft available, are all involved.

55 Natural-draft chain-grate stokers, within the limits of the draft available, produce cheap steam. Low auxiliary power requirements, low maintenance, and controlled loss in the combustion process, make low cost of steam inherent.

56 Natural-draft stokers should be installed:

a Whenever the capacities demanded to meet the station load are within range of the natural draft available

b Whenever the load demand is steady or where peaks can be anticipated sufficiently far ahead to permit building up furnace conditions to meet them

c Within the limits of the above two conditions, whenever induced draft and economizers are used.

57 In Table 1 the author submits some long-period operating figures from chain-grate power stations. While the compiling of such figures by power stations is subject to variations in the methods employed for that purpose, the figures in question nevertheless must be accepted as part of the operating records of large and reliable companies having engineering departments competent to compile such records.

No. 1867*b*

OVERFEED STOKERS OF THE INCLINED TYPE

By Geo. I. Bouton, Detroit, Mich.
Member of the Society

In this paper the author discusses the development and present construction, and method of operation, of the front- and side-feed types of inclined overfeed stokers. Three side-feed stokers are described, with an illustration of a Dutch-oven type of setting. It is stated that 200 per cent of boiler rating can be obtained with 0.3 in. to 0.5 in. draft loss through the fuel bed, and that when forced draft is applied, it should be done with proper regard to air circulation and the maintenance of suction above the fire. Three front-feed stokers are also described, with an illustration of a setting with ignition arch, and an explanation of combustion control by varying the stroke of the coal pusher and the angle of rotation of the bars. The effect of the flux content of ash in determining the fusion temperature is also discussed.

STOKERS of this type are of two general classes, the side-feed double-inclined or V-type, and the front-feed or single-inclined stoker. Development in this country was not hampered by an attempt to fit a square plug in a round hole, and as there were practically no space limitations, it was possible to get the needed combustion space. The early development of these stokers was along natural-draft lines, because that was the draft available and because it was possible to readily burn all the coal required to meet load conditions. As the steam plants have increased in size, the boilers and stokers have also increased in size until this type has apparently reached a limit in a stoker having a projected grate area of 100 to 110 sq. ft. in a single unit, capable of burning sufficient coal, under forced draft, to develop about 2000 b.hp. as a maximum.

2 The double-inclined side-feed stoker was the first to make its appearance in this country, Thomas Murphy fathering this particular line of development, commencing about 1875. His first attempt was hand-operated and was apparently brought about in an attempt to overcome troubles due to cleaning fires and loss of

capacity during cleaning periods. It was placed in the firebox of a marine boiler and had inclined grates, every second grate being movable. It was provided with a revolving toothed cylinder or clinker grinder, extending from front to rear along the bottom of the V, for the removal of clinker and ash.

3 In 1879 Murphy moved this device forward into a Dutch oven, for use in connection with a return-tubular boiler, and added magazines and stoker boxes for feeding coal to the upper end of the grates. He placed a firebrick arch above the entire grate surface, with a second arch enclosing an air space above this arch and with provision for circulating air for combustion over the firebrick arch and through openings in the arch plates immediately above the fire, at the point where the volatile gases are distilled from the coal. About 1880 he provided an engine drive, placing a reciprocating bar across the front of the stoker, with arms and links to operate the stoker shafts which moved the stoker boxes back and forth, the rocker bars which raised and lowered the movable grates, and with a ratchet which rotated the clinker grinder. In all essential features this stoker is the stoker of this class of the present time, as represented by the Murphy, Detroit, and Model stokers, although there are many minor changes in the stokers as built today.

4 About 1885 the front-feed inclined stoker was developed, the Roney and Brightman stokers being brought out at that time. The Brightman stoker was apparently not a financial success and it gradually disappeared; a modified form of it, however, still remains in the Wetzel stoker. The Roney stoker was actively pushed by the Westinghouse interests and is being built by them today. The Wilkinson stoker was brought out by Alfred Wilkinson and was in rather extensive use from 1895 to 1905, particularly for anthracite coal. It has now largely disappeared.

5 Originally the grates in the side-feed stokers were set at an angle of 35 deg. with the horizontal. As the size of the stokers was increased this angle was increased to 40 deg., and later to 45 deg., which is the present standard. Grates at this angle are suitable for bituminous and semi-bituminous coals. With non-coking coals a much flatter grate is desirable, and when coals of this sort are to be burned it is advisable that the stokers be designed particularly for the coal in question. The grates in the Roney stoker are at an angle of 35 deg. with the horizontal, and this stoker will burn

anthracite and bituminous coal and lignite. The grates in the Wetzel stoker are also set at the same angle.

6 Where wood refuse or spent tanbark is used, this is usually burned in connection with bituminous coal, enough coal being burned to keep the grate surface covered. Wood refuse is spouted to the stoker through an opening in the stoker front, or dropped through openings in the arch directly onto the fuel bed. Tanbark is usually mixed with the coal and fed to the stoker in the regular way, that is, through the magazine or hopper. Spent tanbark will run about 60 per cent moisture, and when a half-and-half mixture is used about 85 per cent of the heat supplied to the boiler will be from the coal, the remainder from the tanbark.

SIDE-FEED DOUBLE-INCLINED STOKERS

7 The side-feed double-inclined stoker may have either a flush-front or Dutch-oven setting. The flush-front setting is limited to the smaller return-tubular boilers and horizontally-baffled water-tube boilers. On the larger return-tubular boilers, vertically baffled horizontal water-tube boilers, and boilers of other types it is advisable to place the stoker in a Dutch-oven setting. A Dutch-oven setting materially simplifies the matter of supplying coal to the stoker, and where desirable an eight or ten-hour supply of coal can be stored on top of the stoker. A typical Dutch-oven setting is shown in Figs. 1 and 2.

8 These stokers are made in various sizes, both as to width and depth, varying from a stoker with a grate surface 4 ft. wide and 3 ft. deep to one having a grate 12 ft. wide and 9 ft. deep. In the larger stokers 1 sq. ft. of projected grate area is provided for each $6\frac{1}{4}$ rated horsepower of the boiler, and in the smaller stokers 1 sq. ft. for about each 5 rated horsepower. Where the stokers are set with a Dutch oven they extend in front of the boiler about the depth of the grate surface of the stoker, this being varied to suit local conditions and the details of the boiler. The overall width of the stoker is roughly the width of the boiler setting. The head room required will vary with the width of the stoker. The boiler should be set sufficiently high so that the front header or the sill of the tube-door opening clears the top of the arch about 3 in. This would mean a height of about 6 ft. on a 100-hp. boiler, 7 ft. 9 in. to 8 ft. 3 in. on a 300-hp. boiler, and 11 ft. 3 in. on a 600-hp. boiler. Where vertical water-tube boilers are used, such as

Wickes, Rust, or Erie City vertical, it is advisable to place a combustion space between the rear of the stoker and the tube surface of the boiler. This can be made of somewhat greater cross-section than that of the stoker. The distance from the rear end of the grate surface to the tube surface should be 6 or 7 ft.

9 Stokers of this class are usually driven by a small steam engine operating the reciprocating bar through a chain of gearing,

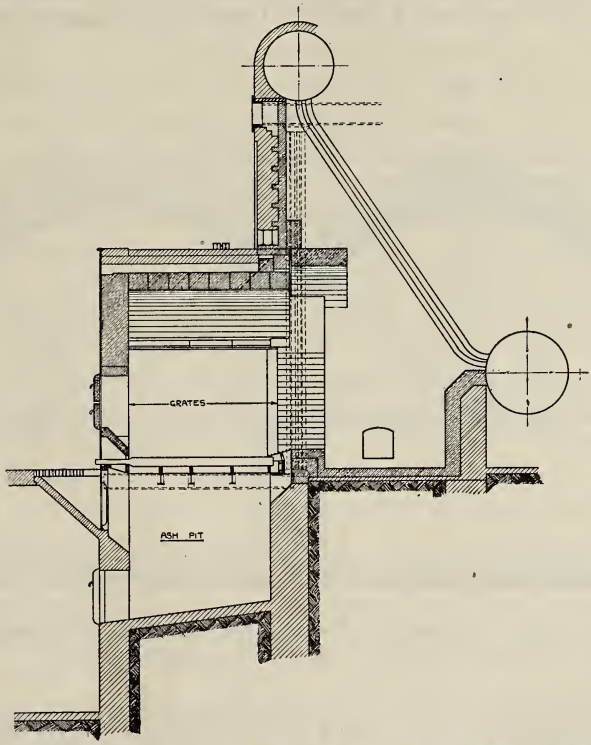


Fig. 1 Typical Dutch-Oven Setting for Side-Feed Double-Inclined Overfeed Stoker under a 500-Hp. Stirling Boiler

the details of the gear reduction being arranged to suit the particular stoker. Where for any reason it is desirable to do so, a motor, either a.c. or d.c., can be used in place of the engine. The speed range of the motor, however, is liable to be considerably smaller than that of an engine: about 3 to 1 for the a.c., and 4 to 1 for the d.c. motor. An engine is preferable to a motor, as the exhaust steam which it supplies is useful under the clinker grinder.

10 As far as the author knows, the most convenient and effective means of chilling clinker so that it can be readily handled is by means of low-pressure steam. This was apparently first employed on a stoker of this class in 1895 and its use is covered by a patent issued to J. H. Nicholson. A pipe was cast in each of the side plates of the grate bearer and holes drilled through the side plate and pipe so that steam was discharged under the clinker grinder and also under the lower ends of the grates. Later on, when a sectional type of grate bearer was devel-

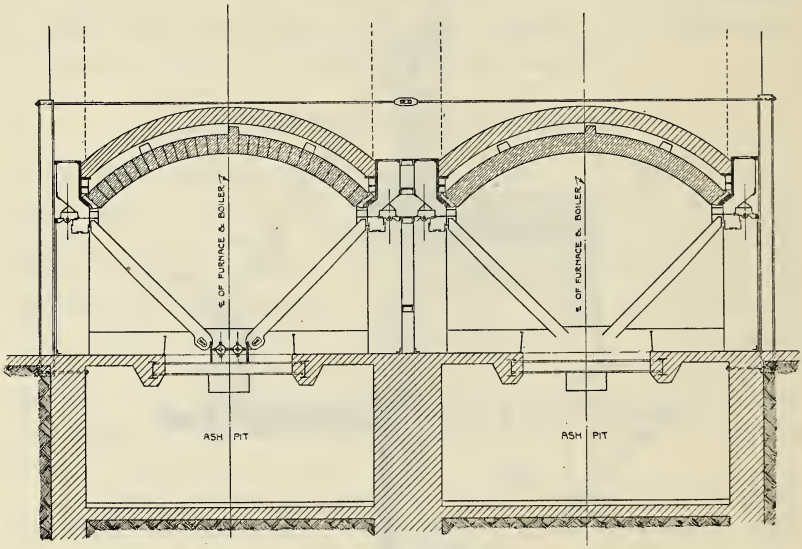


Fig. 2 Typical Dutch-Oven Setting for Side-Feed Double-Inclined Overfeed Stoker under a 500-Hp. Stirling Boiler

oped, a single steam line was placed under the clinker grinder, as this was sufficient with most coals reaching the market ten or fifteen years ago. When the practice grew up of grinding mine dumps and selling the resulting product as high-grade West Virginia coal, the use of additional steam lines under the lower ends of the grates became desirable and these were rather generally used, the result being the equivalent of the earlier design, where steam was discharged under both the clinker grinder and the lower ends of the grates.

11 It is desirable that the fuel be fed as uniformly as pos-

sible to the stoker. The fuel bed is about 8 to 10 in. thick at the upper end of the grates, thinning down gradually toward the clinker grinder. The fuel bed over the lower end of the grates should be thin and active. Minor fluctuations in load should be taken care of by minor variations in the draft over the fire. Where the load changes into a decidedly higher or lower range, the coal feed should be increased or diminished to meet this range, the draft over the fire being regulated to take care of the immediate needs of the boiler.

12 The draft loss through the fuel bed will vary with the amount and kind of coal being burned, the high-volatile coals requiring less draft than the lower-volatile coals. The amount of draft required is also affected by the amount and kind of ash which the coal carries. Where proper draft is provided, stokers of this class will take care of various loads readily up to 200 or 225 per cent of boiler rating, although the draft available usually limits the capacity to 200 per cent of boiler rating or less. Where there is a shortage of draft it is possible to compensate for this in part by increasing the amount of "rousting" or hand manipulation of the fuel bed. At 200 per cent of boiler rating the draft loss through the fuel bed will vary from 0.3 to 0.5 in. water gage, depending on the kind of coal and the amount of rousting.

13 Where it is desired to obtain more capacity than can be obtained with the natural draft available, it is sometimes possible to do this by adding forced draft. Where forced draft is applied to a stoker of this class, it is advisable to see that all the necessary items for its proper installation are taken care of. The air should be admitted to the space beneath the grates, either at the front or the rear, in a line parallel with the clinker grinder; two inlets being used, placed symmetrically with reference to the center line of the stoker. Instances have been found where a blower has been placed through each side sheet, discharging air directly against the grates, and troubles have quickly resulted from such practice. Under natural-draft conditions the amount of air reaching the furnace through the openings in the arch plates increases as the capacity increases, and provision should be made so that this same condition holds good under forced-draft conditions.

14 From time to time small blowers have been applied to stokers of this class without paying any particular attention to the method of installation. The natural draft available is poor,

and the owner wants more capacity and is readily beguiled by a promise of more capacity at a low first cost. A typical case would be one in which there is a draft of about 0.15 in. over the fire under natural-draft conditions. Due to poor breeching design and a dirty boiler, the draft loss between the chimney and furnace is rather large, so that with a 5 or 10 per cent increase in capacity the draft over the fire is reduced to a point where the fire is blown out of the various openings. No attempt is made to insure proper air circulation. Usually these installations are made without referring the matter to the maker of the stokers.

15 The limit of capacity under forced-draft conditions will be that point where the draft over the fire is so reduced that fire is blown out of various openings. This will be when the draft over the fire is reduced below 0.10 in., probably to about 0.05 in. The same conditions hold good here that obtain with forced-draft stokers generally, and the paragraph placed in many contracts for forced-draft stokers requiring the purchaser to provide a draft of 0.1 in. over the fire at maximum rating should be considered to apply properly to installations of this class of stoker when equipped with forced draft.

16 With forced draft properly installed and skilful firing, it is possible under favorable conditions to get in the neighborhood of 300 per cent of boiler rating. The fires are rather thin for forced-draft work and it requires closer attention to the fuel bed to keep the grates covered than in the case of stokers using a much thicker fuel bed.

17 Where these stokers are operated under natural-draft conditions the proper method of regulating the air supply is by adjusting the damper at the outlet of the boiler. This can be done by hand where necessary, and if it is controlled by hand the damper should be connected to a quadrant near the front of the stoker so that it is within convenient reach of the fireman. Where it is controlled in this way the damper will be shifted slightly as the steam pressure rises or falls. There is no objection to using a damper regulator provided this regulator is of proper design. In other words, it should be a regulator which reproduces the action of a skilled fireman as closely as possible and shifts the damper slightly with varying steam pressure. A two-position damper regulator, or one which holds the damper in an open or closed position, depending upon the steam pressure, is worthless.

18 Various forms of automatic control for the draft and coal feed have been used, the favorite form being to vary both the coal feed and the draft as the steam pressure fluctuates. The writer has found no means of determining by the steam pressure whether more coal or less is needed in the stoker. This is a matter which needs the intelligence of a fireman. The automatic device is as apt to be wrong as right.

FRONT-FEED INCLINED STOKERS

19 Of the many forms of front-overfeed inclined stokers which have appeared, the Roney and Wetzel are still being built. The Roney is a natural-draft stoker only. It is not used in conjunction with forced-draft equipment. It is made up of an inclined grate with transverse bars. Each bar is supplied with sectional grate-bar tops so arranged that they allow a uniform distribution of air through the whole grate area. At the upper end of the furnace, flat fuel plates are provided on which the coal can coke, and this prevents sifting of the green coal until the coking process is well under way; by this time there is no further trouble with siftings. There is an agitation given to the grates by a rotary motion on the transverse bars. This motion feeds the refuse and fuel uniformly toward the lower end of the grates. At this point and immediately above the dump grates there is an agitator for the purpose of breaking up clinker that may form in this zone; it also serves the purpose of preventing the avalanching of the fuel bed down the grate while the dump grates are open and the fire is being cleaned. The dump grates are slotted, thus providing active fuel-burning surface in this zone.

20 The angle of rotation of the transverse bars can be varied from nothing to a maximum. The stroke of the coal pusher can also be varied. The ability to control these motions without changing the speed of the engine provides a very wide range of flexibility. It is customary practice to operate Roney-stoker driving units at constant speed, and take up the fluctuation in the rate of feeding coal by means of the variable adjustments on the stroke of the coal pusher and grates. Motors or reciprocating engines are usually supplied for driving Roney stokers. As a rule, constant speed drivers are supplied. The whole stoker-driving mechanism is extremely simple, as a multiplicity of stokers can be driven from one shaft which operates both the feed and grate motion.

Even with this type of drive, every stoker on a lineshaft can be driven at a different rate if so desired. In addition to driving the Roney stoker with an engine, it can also be operated to a limited extent by hand.

21 The thickness of the fuel bed on a Roney stoker is uniform for a given grade of coal and does not vary for different ratings. The best practice is to control the rating on the boiler as desired by means of the boiler damper and not by thickening

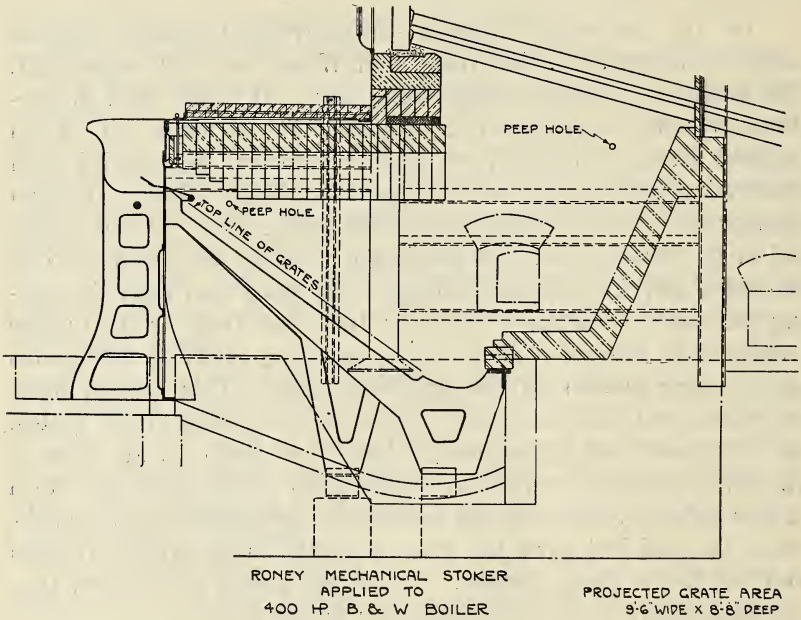


Fig. 3 Front-Feed Overfeed Stoker with Ignition Arch under a 400-Hp. Babcock & Wilcox Boiler

or thinning the fuel bed. There are installations in service where there is automatic regulation controlled by steam pressure, but as a rule it has been found that hand regulation of the boiler damper is about the most reliable and effective method for this class of stoker.

22 The Roney stoker, a typical setting of which is shown in Fig. 3, is applicable to any type of boiler. It is set with an arch to ignite the incoming fuel and burn the volatile gases. This stoker is suitable for burning anthracite and bituminous coals and lignite.

For burning anthracite and lignite a slight change is made in the grate-bar tops so that they will overlap in any position and thus prevent the fuel from sifting through. There are a number of installations where hog fuel has been burned very successfully in conjunction with coal. This is done by using an extension furnace and cutting a hole through the arch so that the hog fuel can be dropped through the arch and on the fuel bed.

23 One of the outstanding features of the Roney stoker is the accessibility of all parts. There is practically nothing that can happen to this stoker that will put it out of service, and it can almost always be operated until it comes out of service for regular cleaning and repairs.

24 The Wetzel stoker is somewhat similar to the Roney, the principal difference being in the form of grate used. Instead of being made up of bars extending across the furnace and rocking, the grate is made up of a series of bars extending from front to rear. These bars are ribbed to prevent sifting, the upper end of the grate is supported on a short rocker arm so that it has a slight forward-and-back motion, and the lower end on an arm placed at a somewhat different angle, so that the motion of the lower end of the grate is practically at right angles to the grate surface.

25 The grates of the Wilkinson stoker were a series of hollow bars roughly rectangular in cross-section. They were stepped on the upper surface and air openings provided in the vertical surface of these steps. Air was supplied to these bars under pressure induced by a small steam jet. Coal was fed to the upper end of the bars and the bars had a slight forward-and-back motion. The bearing for the bars at the lower end was in the form of a box which took care of the accumulation of siftings and permitted their removal when necessary. The stoker was provided with an ignition arch at the upper end and a dump plate or ash pocket at the lower end. It made a rather satisfactory stoker for burning the smaller-sized anthracite coals under boilers of the sizes then in general use.

ASH

26 Most coal ash has a fireclay base with various kinds and amounts of foreign materials which act as fluxes, such as iron, lime, magnesia, and sodium and potassium oxides. An ash con-

taining 40 per cent of fluxes will melt at about 2000 deg. fahr. and one containing 5 per cent at about 3000 deg. fahr. If the percentages of fluxes are plotted in connection with the melting point of the ash, the points will be found to group around a straight line extending between the points mentioned above, the exact figure of course depending somewhat on the relative silica and alumina content of the ash. An ash low in fluxes is therefore desirable because it will be an ash of high fusing temperature and one which can be handled in a stoker more readily than an ash high in fluxes. An ash low in fluxes is also desirable when the maintenance of the stoker and boiler brickwork is considered, as small particles of ash impinge on the brickwork, and the iron, lime, etc., in the ash form an active flux for this firebrick at furnace temperatures.

No. 1867c

THE DESIGN AND OPERATION OF UNDERFEED STOKERS

By H. F. Lawrence, Philadelphia, Pa.
Member of the Society

This paper describes briefly the distinctive features of the single- and multiple-retort types of underfeed stokers, giving illustrations of a number of each type as well as particulars regarding their operation and results that have been obtained. Emphasis is laid on the importance of proper adjustment of the secondary coal feed, to prevent clinker and ash depositing at the lower end of the retorts, and on draft control not too responsive to momentary variations in boiler conditions. Underfeed stokers are said to be particularly adapted to bituminous and semi-bituminous coals, and for carrying sudden overloading. Clinkers are a necessary result of the high temperatures secured, and different means for preventing their adhesion to the side walls are discussed. The relative merits of various fans are considered, and a table of setting heights is given.

UNDERFEED stokers are so designed that coal is fed from beneath the burning fuel. This is accomplished by feeding through retorts with adjustments so that the fuel bed is replenished throughout the length of the retort. The main feed from the coal hopper is accomplished with rams of fixed displacement so that the amount of fuel fed per stroke is a definite amount for a given coal, and therefore the amount fed per hour is accurately controlled by regulating the speed at which the rams are operated.

SINGLE-RETORT STOKERS

2 The first development of this type was the single-retort underfeed stoker. This consists essentially of a horizontal retort, into which fuel is fed from the hopper and distributed throughout the length of the retort. Tuyeres are placed around the edge of the retort and through these air is supplied to the fuel under pressure from a fan. Dead plates or dump plates are placed on each side of the retort, from which the ash and refuse

are removed. For wider furnaces, intermediate inclined movable grates or tuyeres are placed between the retorts and the dump plates. These serve the purpose of providing more grate surface and also of depositing the ash and refuse on the dump plate.

3 Single-retort underfeed stokers do not require large ashpits and ash tunnels below the boiler-room floor. They are cleaned of ash and refuse by dumping into shallow ashpits, which are depressed slightly below the floor line, and withdrawing the ash and refuse through doors in the boiler front at floor level, or by

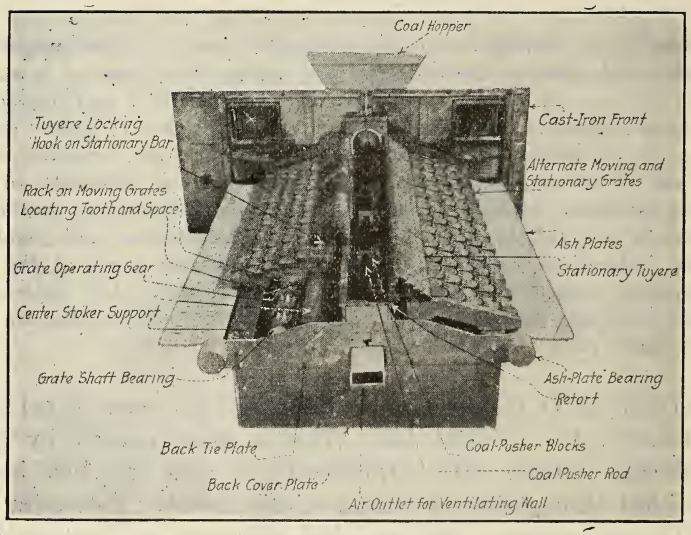


Fig. 1 Sturtevant Single-Retort Underfeed Stoker

withdrawing the ash from the dead plates through doors in the boiler front.

4 These stokers are also particularly adaptable to installations in which more than two boilers are placed in a battery, since side doors are not necessary to their operation. Boiler plants may then secure the advantages of the underfeed type of stoker without the expense of excavation and without being limited as regards arrangement of boilers.

5 Single-retort underfeed stokers are now being manufactured and marketed by five concerns, Figs. 1 to 5, inclusive, showing the various makes and their distinctive features.

MULTIPLE-RETORT STOKERS

6 The multiple-retort stoker is a development of the single-retort and consists of a number of single retorts placed close together and inclined with the ash discharge at the rear. As the coal is burned the ash is formed on top of the fuel bed and is floated to the rear and deposited on dump plates or into crusher pits from which it is readily removed. The continuous ash discharge consists of rotary toothed crushers placed at the rear of the stoker and set low so that a large, deep pit is formed for receiving the ash and burning out the last of the combustible material.

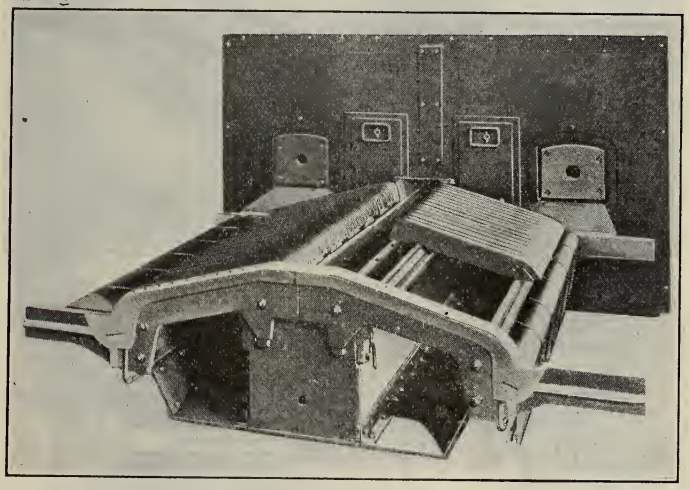


Fig. 2 Type E Single-Retort Underfeed Stoker

7 The secondary coal feed, that is, the feed from the retort to the fuel bed, is obtained in various ways. The Taylor stoker uses additional rams similar to the coal-feeding ram, which are placed in the bottom of the retort. The retort inclination is such that the rams are reciprocated horizontally. The Westinghouse and the new Frederick stokers have a lesser inclination, and the secondary coal feeding is accomplished by large wedge-shaped castings placed in the bottom of the retort. These are reciprocated on an inclination corresponding to the slope of the bottom of the retort.

8 The Jones and Detroit stokers have similarly shaped retorts, and the secondary coal feed is obtained by small wedge-shaped

pushers which are reciprocated horizontally in the bottom of the retort. The Riley stokers accomplish the secondary feeding by

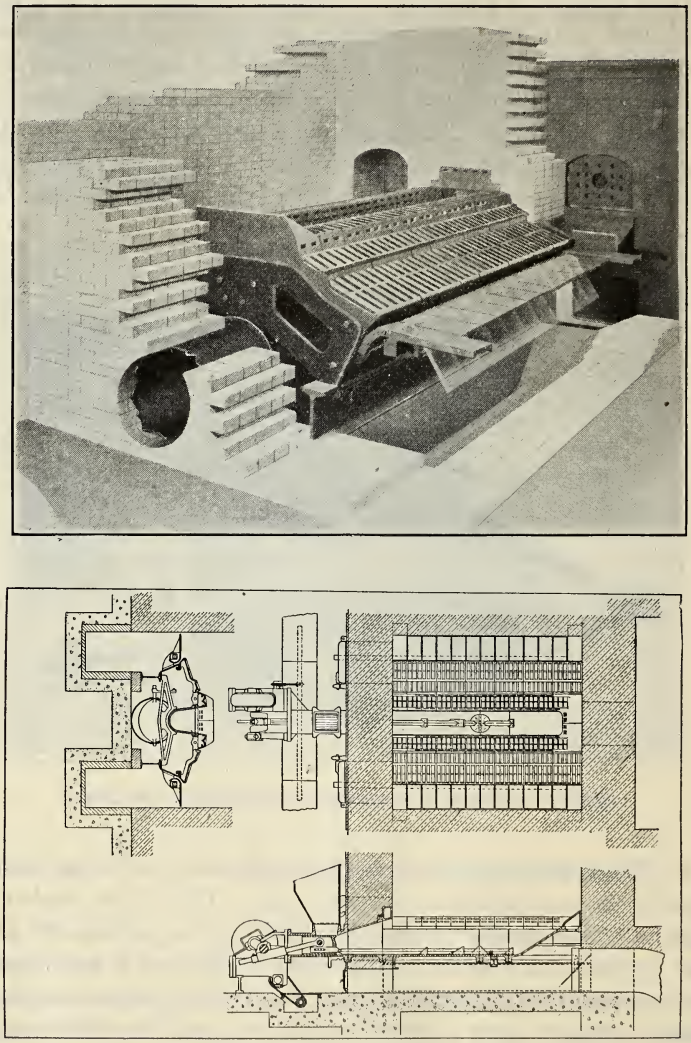


Fig. 3 Detroit Single-Retort Underfeed Stoker

reciprocating the retort sides and the tuyeres. They also vary in the angle at which the tuyeres are placed — from horizontal in the case of the Detroit to 25 deg. in the Taylor stoker.

9 Tuyeres are placed between the retorts, and serve to convert the static head of air into velocity and direct the flow of air through the fuel bed. The tuyere designs naturally are different in each stoker.

10 All designs are for forced draft and cannot be operated at any appreciable capacity with natural draft. The air for combustion is circulated beneath the furnace parts and thereby cools these parts before being discharged through the fuel bed.

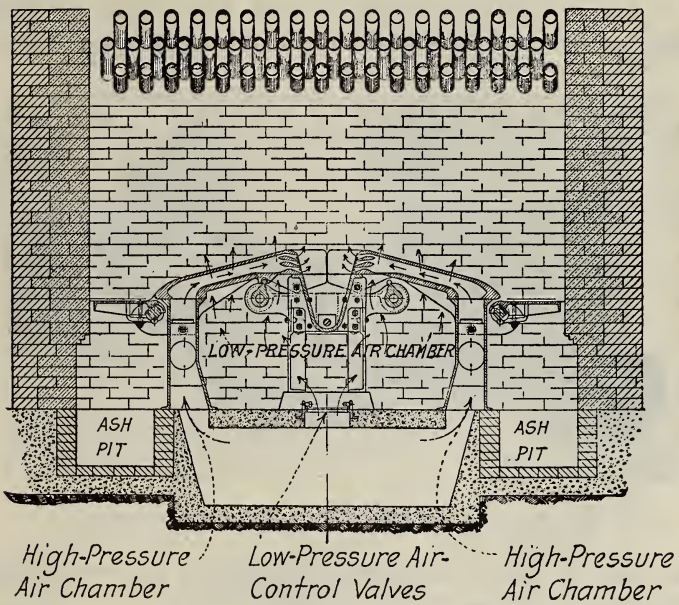


Fig. 4 Roach Single-Retort Underfeed Stoker

11 Figs. 6 to 11, inclusive, show cross-sectional views of the various makes of this type of stoker, and their essential differences in design.

OPERATION OF UNDERFEED STOKERS

12 The operation of underfeed stokers is essentially as follows: The incandescent burning fuel is on top and is replenished throughout the entire retort length from beneath. As the coal emerges from the retort it is coked and spreads over the tuyeres, forming a homogeneous fuel bed across the entire furnace width. As the fuel approaches the surface the volatile matter is completely distilled off

and the fuel completely coked. The surface consists of a layer of incandescent burning coke. The air for combustion is introduced near the point where the fuel emerges from the retort. As the volatile gases are liberated they are thoroughly mixed with air. As the mixture passes up through the fuel bed, higher-temperature zones are reached and complete combustion of the volatile gases takes place when they pass through the white-hot coke at the surface.

13 Smokeless combustion is obtained without the use of special mixing or ignition arches or special brickwork construction. As the fuel bed is replenished from beneath the surface, the burning incandescent coke which is on top is slowly moved toward the dump

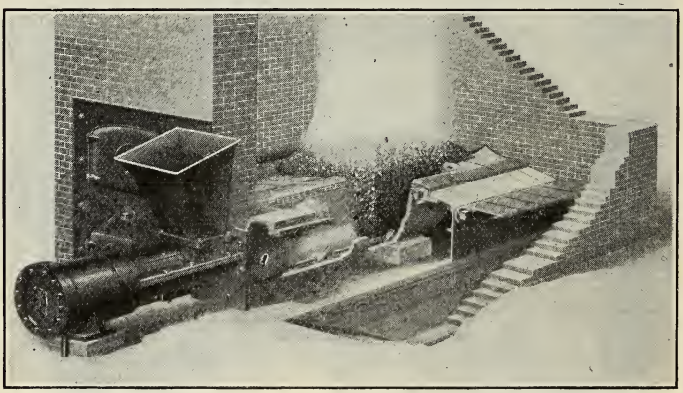


Fig. 5 Jones Single-Retort Underfeed Stoker

plates. As the ash is formed it is floated on the surface and is eventually deposited on the dump plates.

14 The control of the fuel bed is obtained by adjustments of the secondary coal-feeding arrangement. For good operation it is essential that the fuel bed be so controlled that the replenishing coal emerges from the retort through its entire length; the lesser amount being fed from the rear end of the retort.

15 With low-grade western coals more fuel must be discharged from the rear end of the retort than with the high-grade eastern fuels. In general, the greater the quantity of ash in the coal, the longer should be the stroke of the secondary fuel-feeding mechanism.

16 If insufficient coal is fed from the lower or rear end of the retort, the ash, instead of being carried on to the dump plates, is deposited at the lower end of the retorts, and as high fuel-bed

temperatures are always obtained the ash is clinkered and when deposited at this point it blocks the air discharge. After a short interval, fuel from the upper part of the retort, which is not coked, is deposited over this clinker formation and then avalanches on to the dump plates. With this condition of fuel bed it is impossible to secure good results or good operation. With proper strokes of

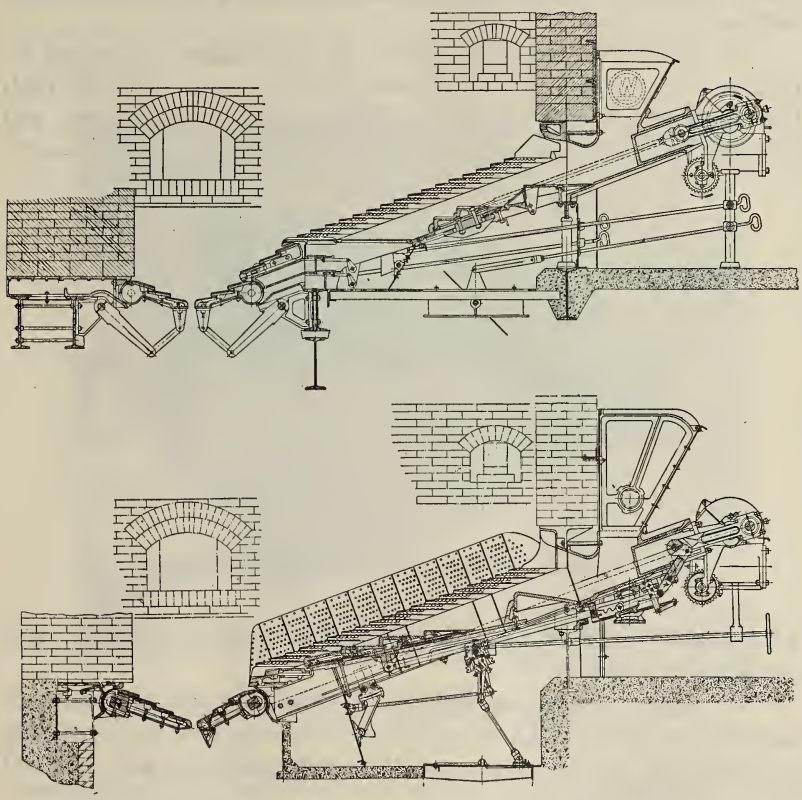


Fig. 6 Westinghouse Multiple-Retort Underfeed Stokers

(Above, standard stoker; below, new model.)

the secondary coal-feeding mechanism this condition can be eliminated, as with the proper amount of coal being fed from this section of the retort the ash and clinker can never be deposited at this point. This is really the important adjustment to be made for various grades of coal, and it probably receives less attention from plant operators than any other variable. When properly adjusted

the fuel bed is automatically maintained clean, and high rates of combustion can be obtained.

17 With the underfeed stoker properly adjusted, fresh fuel will be fed up throughout the full length of the retort. Green fuel moving upward with respect to the tuyeres tends to keep them buried, and consequently the ironwork is in the comparatively cool zone of the fuel bed. For this reason the maintenance is low on underfeed stokers.

18 With the underfeed system of combustion the excess air required can be reduced to a minimum, which means that high fuel-bed temperatures are obtained. The fuel-bed temperature will probably always exceed the ash-fusion point of any of our coals.

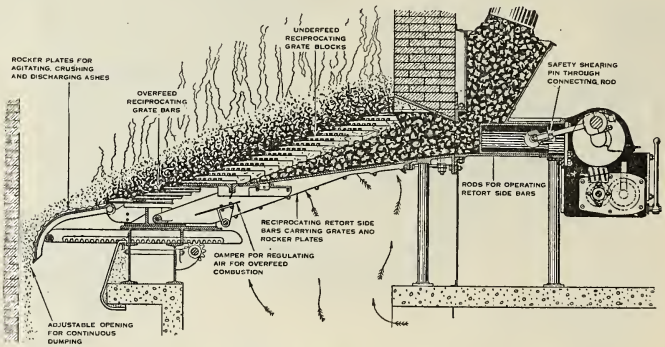


Fig. 7 Riley Multiple-Retort Underfeed Stoker

This means that clinker must be formed in order to secure the best combustion results.

19 These stokers are designed to handle clinker and maintain the fuel bed clean at all times and it is the duty of the operators to keep them so adjusted that they will maintain the fuel bed free of clinkers.

RESULTS SECURED

20 The stokers are particularly adapted for burning bituminous and semi-bituminous coals. However, with only slight modifications, lignites and coke breeze are also burned with excellent results.

21 With the thoroughly coked thick fuel beds carried, this apparatus is very quick in responding to load demands. Under running conditions the boiler capacity can easily be doubled almost

instantaneously. From a banked-fire condition, loads equivalent to 200 per cent of boiler rating and over can be picked up in a few minutes.

22 Standardization and interchangeability of parts have been accomplished to a considerable extent, especially of machined parts, and an effort has been made to simplify and reduce parts necessary to be stocked for replacement. Wherever possible, improvements in design have been so made that the new improved parts will be inter-

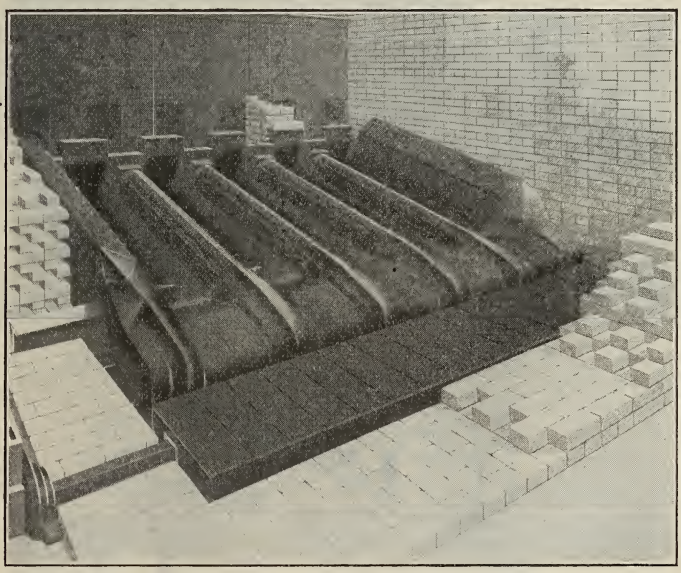


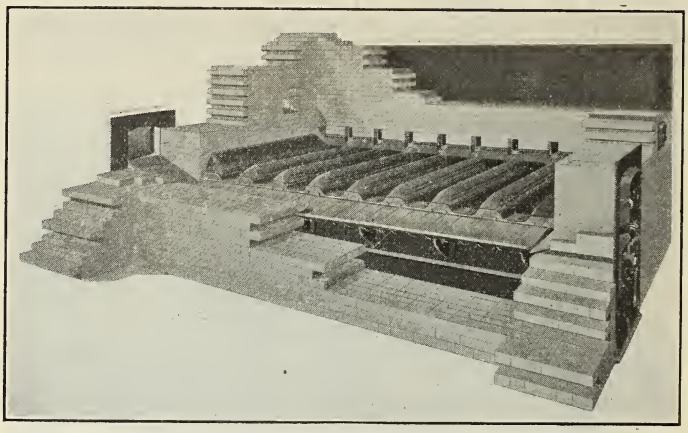
Fig. 8 Jones A-C Multiple-Retort Underfeed Stoker

changeable with the old parts and that no inconvenience will be occasioned by changing from the old to the new. When the new part is adopted it is automatically furnished on all future repair orders.

23 Many times the stoker manufacturer is handicapped by being forced to meet space limitations of the boilers. It is recommended that stokers be selected first, of the proper size and type to obtain the desired results, after which the boiler should be selected to meet the furnace requirements of the stoker.

REGULATION

24 Automatic regulation is receiving a great deal of attention at the present time, and improved equipment has been developed which is giving good results. It is doubtful, however, whether regulating apparatus will ever be developed to the point where adjust-



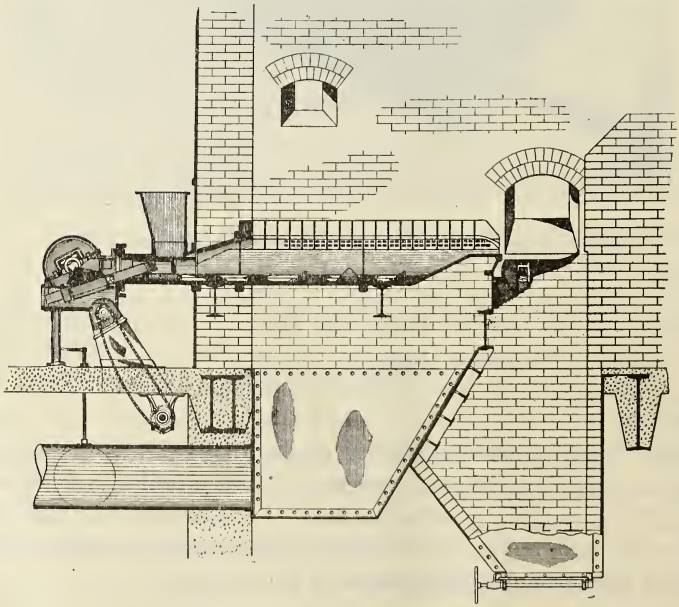


Fig. 9 Detroit Multiple-Retort Underfeed Stoker

ments will not be required from time to time. These adjustments, it may be said, should only be made by an expert observer or fireman.

25 An ideal regulating equipment would be one which controlled all the variable elements in the proper proportion and in exact relation to the load, thereby obtaining a constant steam pressure. Such an equipment would, in addition to controlling the fuel, air supply, and drafts, also be required to control the feedwater in proportion to the load.

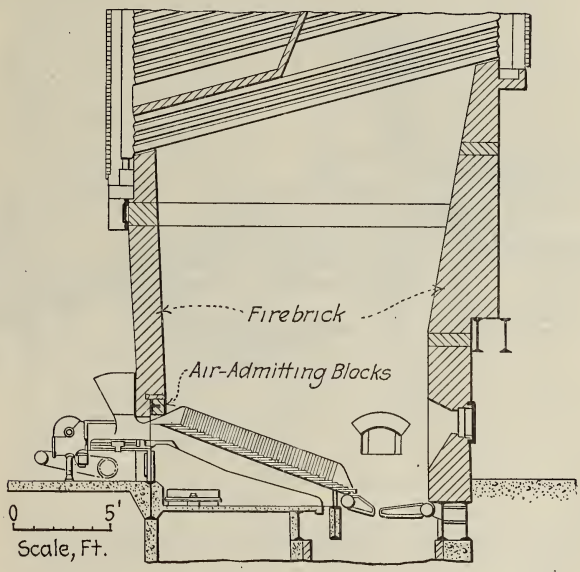


Fig. 10 New Frederick Multiple-Retort Underfeed Stoker

26 Practically all regulating systems in use at the present time are controlled by variations in the steam pressure. The damper regulator is very sensitive to slight changes in steam pressure, and unless its action is retarded by some means, hunting will take place, causing rapid and wide variations in the air pressure. Much better fuel economy is obtained by eliminating the rapid fluctuations in air pressure. This is accomplished by damping the regulator so that a greater steam-pressure variation is required to operate it through its complete stroke.

27 Fig. 12 shows an air-pressure and a steam-pressure chart from a boiler and furnace installation with such an equipment. The

Jones fluid-operated rams are controlled by a Cole automatic valve for each cylinder. This valve is operated from a power source, usually driven from the fan or the fan engine. It is possible to obtain a number of different adjustments for the rate of turning this automatic valve, so that the rate of feeding fuel can be varied for each retort; in fact, all the retorts can be arranged to feed at one speed or at eight different speeds. Furthermore, each valve can be operated by a hand crank, so that the coal can be fed into the retort in very large quantities at any time it is required.

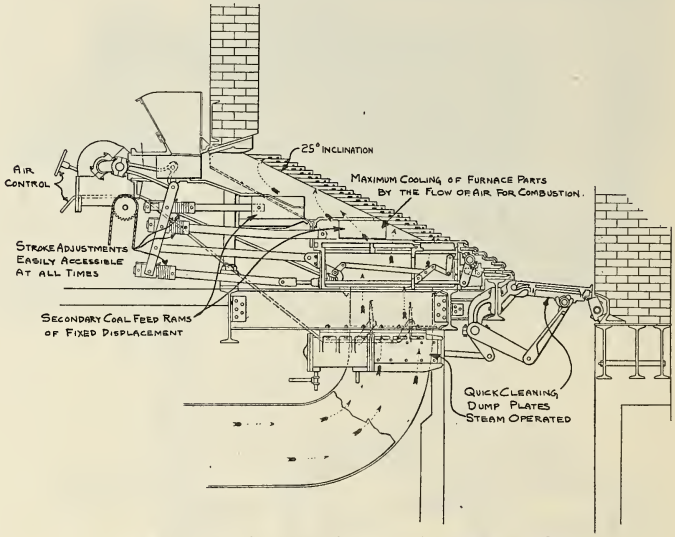


FIG. 11 TAYLOR MULTIPLE-RETORT UNDERFEED STOKER

FANS

28 There are several types of fans used in stoker service, each of which has different characteristics. (See Fig. 13.) The high-speed fan with narrow forward-curved blades (*a*) has a wide range of volume delivered with only slight variations in the static pressure, but is not well adapted for use on underfeed stokers. Due to this pressure characteristic, with two or more of these fans operating together it is practically impossible to keep them in parallel operation. It will be noted that this fan also has a very rapidly increasing horsepower requirement as the static pressure drops and the volume increases. This would only be satisfactory for a motor drive if an exceptionally large motor were used. If for any reason

the pressure should drop, the volume and horsepower would increase so much that the motor would be greatly overloaded.

29 The radial-tip fan (*b*), which is also a narrow-bladed high-

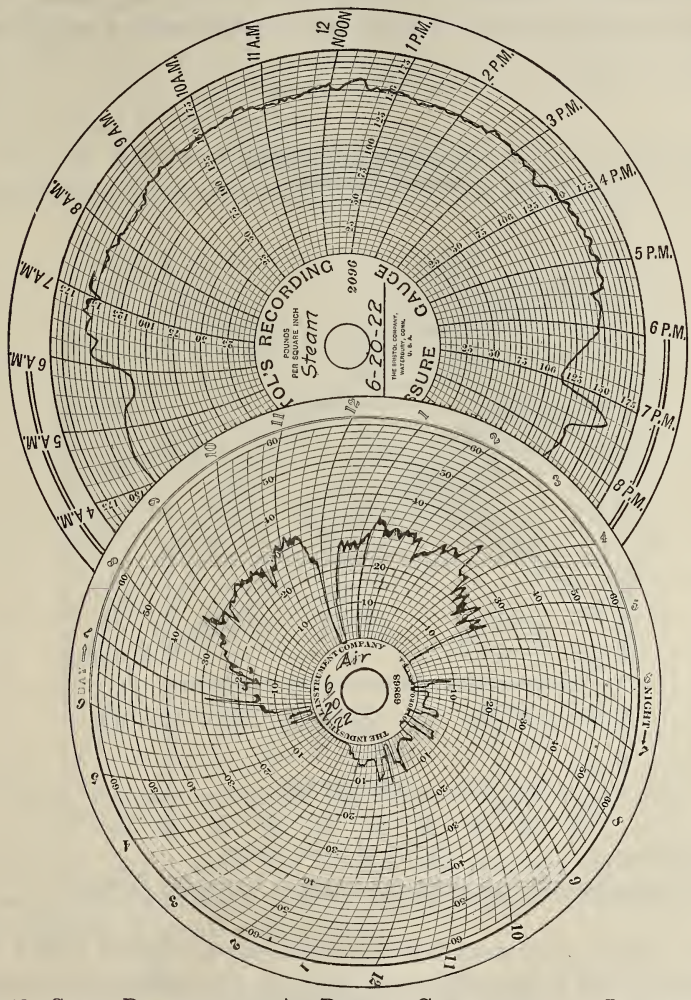


Fig. 12 Steam-Pressure and Air-Pressur Charts from an Installation with Automatic Damper Control

speed fan, has very similar characteristics to the forward-curved-blade fan.

30 The partial-backward-curved fan (*c*) — high speed — and also the steel plate (*d*) — low speed — have characteristics which

are satisfactory, due to a steeper pressure curve. The horsepower increase is not abnormal for a drop in pressure, and these fans will operate in parallel and without trouble.

31 The full-backward-curved fan (*e*) with long blades, which is also a high-speed fan, has the best characteristics for stoker ser-

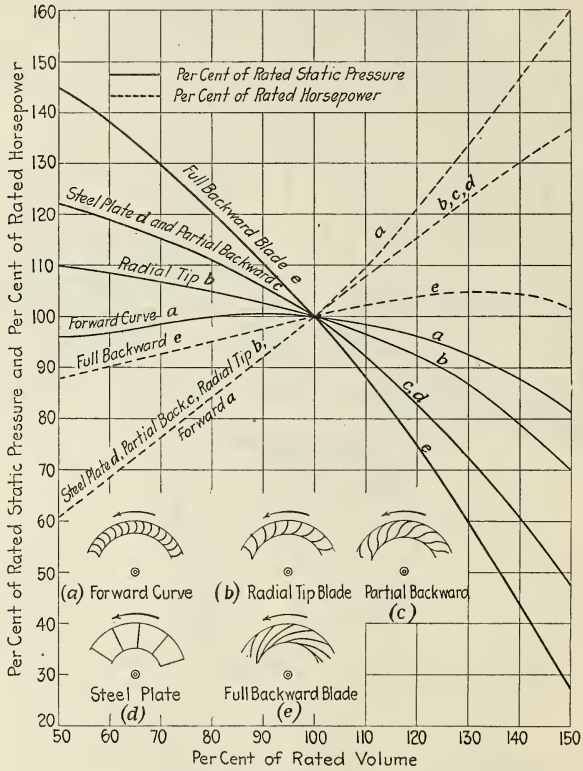


FIG. 13 FAN CHARACTERISTICS AT CONSTANT SPEEDS

vice. It has a very steep static-pressure curve, together with a comparatively flat horsepower curve, and has the additional characteristic that after reaching the maximum horsepower any further increase in volume due to reduction of static pressure, will reduce the horsepower required. The smallest-sized motors can be used safely on this type of fan, and it also has the highest efficiency.

32 The steel-plate fans are usually direct-connected to vertical reciprocating engines. The high-speed fans are usually direct-

connected, when motor driven, while when turbine driven they are usually gear-connected, and sometimes direct-connected, to the turbine.

33 For a fan operating against a constant resistance the power varies as the cube of the speed, the static pressure as the square of the speed, and the volume directly as the speed. In underfeed-stoker practice, however, the resistance is not constant, so that the fans do not follow this law.

34 Fig. 14 shows in the dotted curves the volume and pressure characteristics in accordance with the constant-resistance law, while

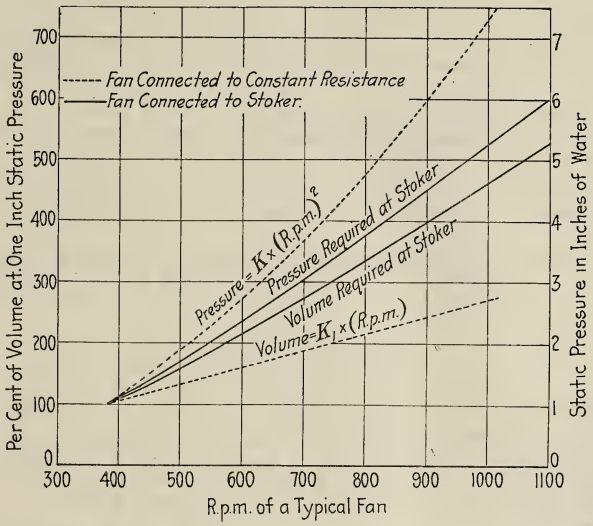


FIG. 14 RELATION OF VOLUME AND PRESSURE OF AIR TO FAN SPEED

the full-line curves show the volume and pressure characteristics at the stoker connection. These curves have been plotted from data obtained in actual stoker tests.

CLINKER PREVENTION

35 The most serious operating difficulties are caused by clinker adhesion to the side walls of the furnace. There are several successful methods for preventing this, the most popular one probably being that in which perforated firebrick blocks are located along the clinker line, through which air is blown from the stoker

air duct. The results from the use of these blocks have been generally satisfactory, although in some instances trouble has been caused by slag clogging the air holes. Care must be used in locating these blocks so that no free air is discharged above the fuel bed.

36 Another method of preventing side-wall clinker uses special high-side tuyeres through which air for combustion is dis-

TABLE 1 SETTING HEIGHTS FOR VARIOUS TYPES OF BOILERS EQUIPPED WITH STOKERS

(Min. = absolute minimum; P.M. = preferred minimum, i.e., the minimum heights recommended.)

Type of Boiler	Type of Stoker to be Installed													
	Multiple-Retort Underfeed		Single-Retort Underfeed				Side Overfeed		Front Overfeed		Chain Grates			
	Taylor, Westinghouse, Riley, Jones A-C, Detroit		Type E		Jones Single-Retort, Detroit Single-Retort		Murphy, Detroit		Roney		Natural Draft		Forced Draft	
	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.	Min.	P.M.
Water-Tube:														
Horizontal	10′	12′	10′	12′	8′	10′	8′	11′	8′	10′	10′	12′	12′	14′
Inclined (Hor. M.D.)	7′	8′	6′	8′	6′	8′	5′	7′	6′	8′	6′	8′	7′	8′
Inclined (Vert. M.D.)	5′	6′	5′	6′	3′6″	5′	3′6″	5′	3′6″	5′	3′6″	5′	6′	8′
Vertical (Hor. M.D.)	3′	4′	3′	4′	3′	4′	3′	4′	3′	4′	3′	4′	3′	4′
Vertical (Vert. M.D.)														
150-hp.	4′6″	5′	4′6″	5′	4′6″	5′	3′3″	....	3′6″	4′6″	4′1″	4′7″	5′	5′ 6″
250-hp.	5′6″	6′	5′6″	6′	5′6″	6′	3′3″	....	3′6″	4′6″	4′1″	4′7″	5′	5′ 6″
500-hp.	6′	6′6″	6′	6′6″	6′	6′6″	3′3″	....	3′6″	4′6″	4′1″	4′7″	6′	6′ 6″
Horizontal Return Tubular:														
72-in.	8′	10′	8′	10′	7′	10′	7′	8′	6′	8′	7′	8′	8′	10′
84-in.	8′	10′	8′	10′	7′	10′	7′	9′	6′	8′	7′	8′	8′	10′

charged, these tuyeres extending high enough along the side walls to prevent the clinker adhesion to the brickwork.

37 Still another method which has been very successfully applied in a number of plants, is one in which cast-iron side-wall air boxes are used. These are independent of the stoker structure, except through the air connections. The faces of the boxes toward

the fuel bed are made of small overlapping ribbed plates. These plates are solid so that no air is discharged through them. Air enters one end of the box from the stoker air chamber and is discharged under the tuyeres from the other end of the box; this air circulation being sufficient to prevent burning of the plates and the adhesion of clinker. These boxes should not be less than 10 in. wide.

38 Carborundum bricks are satisfactory for this purpose with

TABLE 2 DEFINITIONS OF SETTING HEIGHTS FOR VARIOUS TYPES OF BOILERS

Type of Boiler	Setting Height
Water-tube, horizontal	Floor line to bottom of header above stoker
Water-tube, inclined	Horizontal mud drum: floor line to center of mud drum Vertical mud drum: floor line to top of mud drum
Water-tube, vertical	Horizontal mud drum: floor line to center of mud drum Vertical mud drum: floor line to top of mud drum
Horizontal return tubular	Floor line to under side of shell

some coals. However, when the ash contains much iron, carborundum brick is rapidly eaten away.

SETTING HEIGHTS

39 The Stoker Manufacturers' Association, in conjunction with the American Boiler Manufacturers' Association, have adopted minimum setting heights for all types of boilers which are given in Table 1. Setting heights for the different types of boilers are defined in Table 2.

40 A number of recent large boiler units have been set considerably higher than as specified in Table 1. The boilers of the new Hell Gate power station are set 21 ft., giving a furnace volume of 16½ cu. ft. per sq. ft. of grate surface, or 390 cu. ft. of furnace volume per 1000 cu. ft. of boiler heating surface.

41 Frequently the combustion space required is stated as a function of the coal-burning capacity. This is misleading since it leaves out of consideration all conditions imposed by the arrangement of the boiler baffles and stoker in relation to one another.

42 It is desirable to keep the velocity of the rising gases in the furnace as low as possible, but a larger horizontal furnace section, with consequent large volume, may not necessarily do this.

43 Ample height of the boiler above the stoker should be secured in order that combustion of the gases may be completed before they enter the tubes.

44 High setting heights impose a more severe service upon the brickwork. Extreme care must be used in designing the furnace walls, so that they will stand not only the high furnace temperatures, but also the load. It is common practice at the present time to expose all of the first few rows of boiler tubes to the radiant heat from the fuel bed. This gives lower furnace temperature and greater life to the brickwork.

45 No arch construction or special brickwork is required in the application of underfeed stokers, and in fact it is preferable not to have arches. On account of the differences in the coefficients of expansion of different kinds of brick; however, only one kind should be used in the furnace.

46 Proper provision must be made for taking care of the expansion as the setting heats up. The brick should be carefully sized so that thin joints can be obtained. Each brick should be dipped in a thin fireclay wash and tapped into place with a wooden mallet until it touches the bricks next to it.

47 Walls should never be so constructed that they overhang or lean toward the furnace. Walls which slope outwardly from the furnace will give much longer service.

DISCUSSION [1]

A. H. Blackburn. In drawing attention to the many types of underfeed stokers, Mr. Lawrence has omitted to mention the lateral-retort stoker brought out by The Under-Feed Stoker Company of America during the present year.

The novel feature of this stoker is that it feeds the coal in through a main retort extending from the front wall of the furnace to the bridge wall, with lateral retorts branching off the main or central retort at right angles as shown in Fig. 1.

The coal from the hopper is fed by a steam cylinder into the central retort, and the coal from this central retort is forced into each lateral retort by means of laterally operating pusher blocks operated by an auxiliary steam cylinder. The length and

[1] Applies to papers Nos. 1867 *a*, *b*, and *c*.

the timing of the stroke of these lateral pusher blocks may be varied and separately controlled, as the auxiliary cylinder is independent of the main cylinder.

This stoker is the same in principle as the multiple retort stokers, being practically two multiple-retort stokers placed back to back, fed by a main retort and dumping at the sides of the boiler. The coal is fed up underneath the entire fuel bed throughout the various lateral retorts, with a gradual rearward and upward progress of the coal. This is the only high-duty, side-cleaning stoker on the market in which the underfeed principle is wholly embodied.

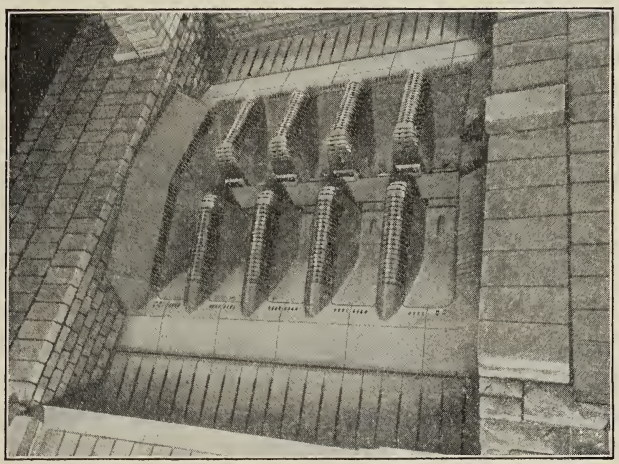


Fig. 1 Top View of Stoker Showing Arrangement of Five Lateral Retorts and One Main Retort

The number of strokes of the feed cylinders are controlled by two Jones automatic valves, which valves are regulated by the pressure of steam, so that the coal feed is in proportion to the demand for steam.

The front and back walls of the furnace are protected by high air-cooled tuyeres, the stokers having been designed for working continuously around 200 per cent boiler rating. The design of the stoker enables it to be installed without a basement and under low-set boilers that may be already installed.

J. M. Drabelle. In the central West, particularly in Iowa, there is a rather peculiar type of fuel available, namely low in

heat value running 8500 to 9000 B.t.u. per pound as screenings. The ash content is very high, 20 to 35 per cent, and the ash easily fusible. To burn this certain definite furnace conditions must be met, namely an undisturbed fuel bed, a large furnace volume and a high ignition rate.

The chain-grate stoker, judging from several years of actual operating experience, is the only successful stoker for handling this type of fuel. Every stoker has its field of application and for high ash Western coals as above mentioned it is the only stoker that meets the requirements above stated.

Very considerable work has been done in connection with this problem working with Mr. Marsh. We have found the following requirements have to be met:

Proper arch construction

Large furnace volume

Definite direction of leakage air coming over the back end of the grate in order to reduce the carbon in the ash to the lowest possible value. This slight air leakage also furnishes additional secondary air for the completion of combustion of furnace gases.

In judging stoker performance it must be borne in mind that it is the total cost per thousand pounds of water evaporated and not percentage efficiency that must be dealt with. The maintenance problem is a serious and important one, also the amount of operating labor required. While some stokers may develop higher efficiencies, when the amount of auxiliary power required for the operation of fans and other special devices is deducted they do not present any real gain in efficiency on Iowa coal. The item of investment must also be borne in mind for this in turn determines fixed charges.

We have found over many years' experience that the chain-grate stoker gives sufficient flexibility in handling loads to take care of conditions in power plants in this state. For example, in one large railway power plant the chain-grate is successfully handling a widely fluctuating load made up of a combination of demands from an extensive street railway system and a large interurban network and the boiler efficiency in this plant runs for over the month well around 73 per cent.

The ideal chain-grate stoker, to my mind, would be a com-

bined natural-draft and forced-draft stoker, operating under the normal conditions of the station on natural draft and over the peak load with forced draft, thereby reducing the amount of auxiliary power required to the absolute minimum, resulting in a lower cost per thousand pounds of water evaporated.

Walter N. Polakov. Quite apart from the excellence of design and construction of a mechanism, and altogether independent from it adaptability to fuel used, the success or failure of a stoker largely depends upon the mode of its use.

Sometimes it is claimed that a mechanical stoker is an automatic device and that by the virtue of its so-called automatic action the plant owner can " forget his power plant." It is only too often that this lullaby has its effect with telling results of shameless fuel waste.

A stoker is automatic only in so far as it replaces a certain amount of physical exertion by the substitution of mechanical power. For this reason alone the stoker attendant is no longer interested in how he can spare himself the pain of shoveling unnecessary coal into a furnace, and unless he has other stimuli he will let the stoker feed as much coal as it can and make as little steam as it may. Again, when *supplementing* a mechanical stoker, some automatic regulating devices are introduced, which as a rule control only one or a few detached factors, but never *the operation as a whole.* Worse yet, in a large number of cases, these automatic controllers are working *either a little behind the time* or make adjustments of conditions by *steps.* In both cases the consequences are that losses are increased coming and going. Even with oil firing I have on record cases where automatic regulators of merit had to be discontinued and manual control introduced in order to increase the evaporation.

In other words, no machine, however perfect, however automatic in its function, can replace the human intellect. On the other hand, there is a grave question in my mind whether it should, for, if we take from the operator the mental stimulus of thinking while he is working; if we will make him a mechanical adjunct to an automatic machine, we will invite trouble by trusting this vital process in industry not to man but to automatons.

What I have said is neither theory nor generalization. It is a summary of facts which came under my observation during the

last score of years without a single exception. A few illustrations may be to the point:

Not many years ago, a public utility plant in central Pennsylvania (Warrior Ridge, Penn Central Light & Power Company) had hand-fired boilers with well-trained help and a reasonable array of instruments helping them in the intelligent performance of work. Under the stress of war conditions, the management installed, after raising boilers to suitable height, a certain approved type of stokers. What were the results? The average of three years of hand firing indicates the combined boiler efficiency of 72 per cent, the performance of the stoker on the acceptance test was higher than that, but in the daily operation a year after it averaged only something like 65 per cent.

Again, I may recall the case of a large manufacturing concern (The Celluloid Company) where mechanical stokers were working for several years at an efficiency averaging between 50 and 60 per cent. Stoker company experts and engineering construction companies were called in from time to time, but the results under the traditional form of management in the power house were never lasting enough to influence yearly, or even monthly averages. At last a change has been made, not in the equipment, but in the mental attitude of management, which resulted in the training of personnel, arousing intelligence in boiler room, in providing mental stimulus, with the outcome that the force dormant in human nature, the " lure of perfection," led them to competition with their own past. The efficiency of 50 to 60 per cent became a matter of sad memory, and during the last three or four years the combined stoker and boiler efficiency averaged 74 per cent.

There is still another type of cases which may be exemplified by an instance of a plant in New England. A well-known type of stoker was employed there under vertical boilers (at Lewiston Bleachery & Dye Works), and while the efficiency was as a rule slightly over 50 per cent, the nuisance of smoke and shortage of capacity led them to investigate the troubles. The same well-beaten path was followed; instruments were put in, man's interest aroused, instruction and training provided, stimuli created. Then it was found out that the load which eleven boilers were unable to carry under the old régime was carried safely and economically by only seven boilers. No smoke complaint, no clinker trouble, and the efficiency on the annual average showed a little better than 72 per cent.

The lesson to be drawn is this:

1 The best stoker is one which permits the largest amplitude of manual control
2 The best results can be secured from a stoker *only* when the management assumes responsibility, and
   a Does away with flapper-engineering that installs stoker without full assortment of instruments
   b Introduces intelligence into boiler rooms by training personnel through a long series of experiments, demonstrations and explanations, and
   c Elevates the man behind the stoker from the position of an automaton to a truly human class, where the lure of perfection reigns supreme.

J. R. Fortune. There should be some way of designating stokers other than by the terms "underfeed" and "overfeed." There is only one type of stoker that, in my opinion, can truly be described as overfed, and that is the sprinkling stoker, which is not very well known in this country but which is quite commonly used abroad. This is the only stoker in which the fuel is fed on top of the burning fire. The fuel in the stokers of the Murphy, Roney, and chain-grate types is always, in effect, underfed. In other words, there is a layer of fuel unignited close to the grate bars near the stoker hopper, and this is overlaid with fuel which is burning almost all the way to the feed opening of the stoker. The unignited fuel close to the grate bar may extend halfway down the grate.

It might be said that an underfed stoker is a stoker in which the fuel is fed beneath the burning fire, and under the air-feeding devices of the stoker. If we use this definition, the inclined multiple-retort stokers would not comply as the fuel is fed horizontally and practically parallel to the openings of the tuyeres.

If the design of the Murphy stoker is studied it is seen that the fuel is fed underneath some of the air-feed openings of the furnace, and it is a fact that some of the earlier designs of this stoker were provided with vertical tuyeres located in the feed openings as shown by Fig. 2 which discharged air into the furnace in a precisely similar manner to the method now used in the multiple retort stokers.

I suggest, therefore, that it would be a good plan to drop the designation of "overfeed" and "underfeed" from the mechanical stoker vocabulary.

Alex. D. Bailey. Mr Marsh's statements that "modern steam turbines called for greater capacities from boilers" is not the whole story. Turbine development has been in a way parallel with boiler development, and along with the development of these two major pieces of equipment has been the coincident development of all plant equipment. So far as the boilers themselves are con-

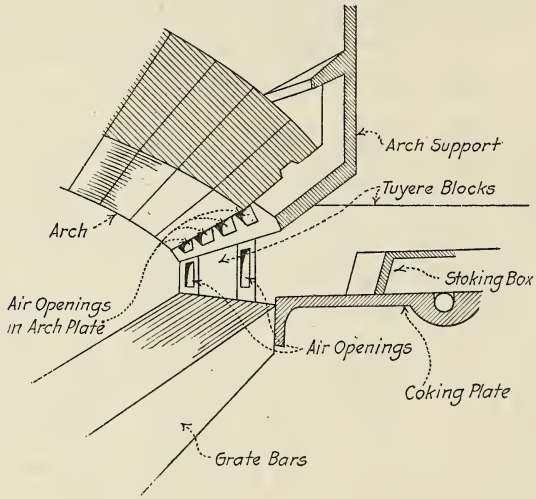


Fig. 2 Tuyere Blocks Located in the Feed Opening of the Murphy Stoker

cerned, the demand for higher capacities has been occasioned by increased fuel costs and increased equipment costs, the latter due partly to higher pressures and partly to economic conditions.

That the boiler proper has shown itself capable of taking care of these rates of heat absorption without materially decreasing its efficiency is evidenced by the very marked development of stokers and other coal-burning equipment during the past few years. At no time in the past has the improvement been so marked or so rapid. The chain grate, which is probably one of the oldest forms of mechanical stoker known, has had to fulfill its part in this development in order to justify its existence, and the development of the forced-draft chain grate is the result.

For higher boiler capacities, increased gas volumes are essential, which means higher coal-burning capacities and rates. As the rate of coal-burning has definite limits, as has been proved, increases have been necessary in grate areas, resulting in the so-called over-stokering of boiler installations. Additional gas volume has also required additional furnace volume in order to assure proper combustion conditions. All this in turn has increased the investment cost of this equipment, which, together with the increased cost of fuel, has not only justified but made necessary the various refinements in stoker and furnace design by which maximum efficiencies are obtainable.

Just as the natural-draft chain-grate stoker has made possible the use of coals which were unsuited to existing types of stokers which were consequently cheap, so the forced-draft chain grate has made available not only low-grade fuels, but also coke breeze and smaller sizes of anthracite, which had not been previously considered suitable for power production. This economic saving alone justifies the improvement in chain grates.

Wm. R. Roney. We cannot say that the power users are responsible for all the troubles that arise in the boiler room, and we cannot say that the stoker builders are responsible for all. It is a question of united responsibility and the details of the burning of the coal, and how it is handled, be the power house large or small, are details of engineering outside of stoker-making. I feel that it is a question of responsibility, well-divided all the way from the humble fireman who shovels the ashes out of the ashpit, to the engineer who sits in his office and directs. Consequently we are — I say we, because I am not now in the stoker business, but as a stoker advocate I say we are responsible for a large part of what goes on in the boiler room and as the time goes on, we will be more and more responsible.

From my own experience in trying to make a stoker work against prejudice, ignorance, bad handling, bad fuel and bad conditions, I learned a good many things; but principally that the human element that handles the stoker, that handles the boiler-room, that handles and develops the power, is the most important of all.

David Moffat Myers. I would like first to supplement what Mr. Roney said with regard to the human element. I would like

to put his remarks in engineering terms, and that can be done very simply. The efficiency of any process is equal to the efficiency of the mechanical equipment, multiplied by the efficiency of the human equipment.

I have five questions to ask.

1 In the underfeed stoker the coal is gradually elevated to the top zone of the fire bed, where it is subjected to the highest temperature of the fire which promotes fusion and clinker formation. What can be done about it?

2 We have been dealing with steam production on land. Approximately 67 per cent of all the coal mined goes to making steam on land. The bunker coal is a very small percentage, but these stoker builders seem very careful to avoid getting into the trouble of applying stokers to production of steam in vessels on the sea. It is a tough problem, but why should they avoid it?

3 I would like to hear the stoker manufacturers who are here today, tell me plainly what they think about automatic regulation. I have, when buying stokers, put this problem up to them: Here are people who promise to add 2 or 3 or 4 per cent to the efficiency of the stoker installation if you will install their system of automatic regulation. I have gone back to the stoker people from whom I am buying the stoker equipment and have told them that if they will add 1 per cent to the stoker guarantees, I will buy their equipment. They throw up their hands and are not interested. Why?

4 Mr. Lawrence gives a very interesting table showing the height of boiler setting recommended by the Stoker Association. Unfortunately, he does not relate those figures to the capacity rating expected from the boilers. The combustion space is a direct function of the rating from the boilers with a given fuel. The more coal burned per minute, the greater amount of combustion space necessary in order that a molecule of the fuel may remain a given length of time in that combustion space for complete combustion. Why not relate the height of boiler setting to boiler rating, or at least to pounds of coal per hour, for different kinds of coal?

5 Mr. Marsh gives a table of records obtained with chain-grate stokers. I am unable to find in that table any results of the burning of the fine sizes of anthracite which I would like to have for comparison with the results in other plants under my observation.

J. E. Woodwell. There is a stoker for every class of coal, and a class of coal for every stoker. That is to say, there is a certain furnace volume, rate of feed, and ash-fusing temperature which depend on definite qualities of the coal, and which determine the design of the stoker. So it is not a competition between the chaingrate and the underfeed or the overfeed, or the inclined-feed stoker for every location and condition. They all have their field, they all have their limitations. Recently we have discovered some valuable features of stoker design. We are just beginning to find out that one of the principal essentials of stoker design is furnace volume, and we are learning that when a boiler manufacturer or stoker manufacturer advises 300 to 400 per cent rating on peak, the owner of the boiler must expect an increase in replacement of furnace linings, etc., and have his boiler laid off for repairs. Therefore, I am an advocate of excess stoker capacity just as a reserve against frequent furnace repairs so that the boiler will be in service nearly 100 per cent of the time.

In regard to automatic control, I do not agree with Mr. Polakov. In our last station there is a system of automatic control which coördinates the draft fans, the stoker motors and the outlet damper of the boiler. The control is by steam pressure.

Manual control can, in my opinion, never equal automatic control under proper supervision.

Theodore Maynz. One point in the chain-grate stoker that we operators have to watch is the carbon in the ash. With some coals we can run with 35 or 40 per cent excess air and get good carbon, and we have to raise our air to 60 or 70 per cent with another coal before our total loss is at the minimum.

Another point is uniformity in the size of coal. In a stoker 12 or 13 feet wide, if the coal is not absolutely uniform, we will get coarse coal, as a rule, on the sides and that means excess air and a bad fuel bed. Therefore, with chain-grate stokers it is important to have the coal as uniform as possible. This is not easy. We can mix it up in the bunkers and by the time it gets to the coal hoppers, it is again segregated, the coarse going through the center and down the sides while the fine coals are packed about a foot from the wall to within a foot of the center. To obviate this trouble we are trying, and have had some success with, tilting and split gates.

Another point is the seal at the rear end of the stoker. This can be sealed in two ways. One way to seal it is by a bank of coal against the water back that cuts off a lot of the grate surface and seems to give a much higher carbon in the ash. Another method is an internal seal. We also seal it by means of ashes between the lower portion of the grate and the ashpit. If we run with too high a bank in the rear, that is, over 3 or 4 inches, we cannot get the right amount of carbon in the ash.

With a forced-draft chain-grate stoker, the dirt coming out from the front is a big objection. We have not been able, nor have the manufacturers of our stokers, been able to seal the front part of our stoker, and it makes almost an unlivable boiler-room.

I agree with Mr. Polakov that the stoker operator should be given proper instruments. They are absolutely necessary and the men appreciate them.

Stoker operators would like stoker manufacturers to solve the following problems:

a To get a stoker that will burn practically all types of coal with uniform efficiency
b The elimination of clinkers up the stack and slag on the lower tubes of the boiler
c A clean boiler-room with stoker-fired boilers
d Stokers with reliabilities of medium-sized turbines.

Edwin Lundgren. I would like to answer some of Mr. Myers' questions. The reasons stokers have not been applied universally to marine practice are, first, that stokers must take coal from almost any port and hence very different kinds of fuels which may or may not burn satisfactorily. Another difficulty is the low space that is provided between the tubes of the boiler and the bottom of the ship, which makes it very difficult to install any but the horizontal type of stoker.

However, The Combustion Engineering Corporation have applied their type E stoker to several ships in Europe, and in most cases successful operation has been reported. There have been difficulties, I believe, with certain types of fuels, but the operators seem to think that the stoker can be made satisfactory for marine service.

Mr. Myers spoke about automatic regulation. It has been my personal experience that the best automatic control is a really

intelligent fireman to whom has been given the simple proper instruments which will guide him in the operation of the stoker.

Of course, there are industrial plants which cannot afford this high-grade operator. In such cases, perhaps, automatic control with a hit-or-miss operation at the best, will give most satisfactory operation.

Albert A. Cary. In the summary of Mr. Marsh's paper, a rate of combustion of 40 to 45 lb. per sq. ft. of grate per hr. is mentioned, and in Par 47, 55 to 60 lb. per sq. ft. per hr. Such statements mean practically nothing. In a previous paper I stated that when a fuel carrying combustible gaseous matter is charged upon a hot fuel bed the gaseous matter is distilled and rises above the grate into the combustion chamber where it is burned. It is only the coke or fixed carbon that remains behind and is really consumed on the grate.

If we want to specify the capacity of a stoker by pounds burned per square foot of grate, it is practical to specify the pounds of fixed carbon per square foot of grate. There is no difficulty in burning high volatile coal at these higher rates of combustion, but there is difficulty with Georgia Creek or Cumberland.

One of the great troubles not spoken of in the paper on chain-grate stokers, and which has gradually been overcome, was the burning of the caking coals of the East. The difficulty was to get the coal on the grates, and getting it in a condition of non-caking coal such as occurs with the western coals.

The "rousting" and hand manipulation of overfeed stokers, spoken of by Mr. Bouton in Par. 12 of his paper, is the curse of this type of stoker. There is less of it done now than before, but it should be done away with in order to get the best results.

Alfred B. Carhart. It has been mentioned in the discussion that automatic control can maintain an average condition but only with the intelligent operator can adjustment be made to varying conditions so as to control the actual results of the moment, rather than making historical records to show what mistakes were made. We must have a guide for present operation, not merely a distant record in the office of the superintendent of the works, which shows the errors that have been made.

I think that too often the manufacturers of equipment, whether boilers or stokers, or any other automatic device, have made us believe that with the installation of such machinery, we could greatly increase our efficiency. No doubt it is so, and it is not a criticism, but rather a commendation on stokers, that their installation means not only greater combustion of coal, but also a double opportunity for loss and waste, which is not always properly recognized at the present time. We often wake up to it afterward in the actual operation.

We should take into account that having such a valuable piece of apparatus in our hands, we should take pains to see that it is intelligently used rather than believe that by the installation of such machinery we have rid ourselves of all future thought and trouble. There are still intelligent operators to be had in this country, and we do not have to depend upon the lumper and helper type, which is so often looked upon as the ideal labor in the operation of a stoker.

C. G. Spencer. This symposium has brought out very clearly the fact which we all knew, that each type of fuel must have a stoker adapted to it, and the furnace must be designed for that type only. Ever since war conditions came upon us, we have had brought to our attention that fact that we cannot get the type of fuel that we would like at all times. Instead of having, as Professor Breckenridge has pointed out, five types of stokers to burn 75 types of fuels, we need one type of stoker which will burn 75 types of fuel. This is the big problem ahead of fuel engineers at the present time.

H. G. Heaton For many years power station operators have experienced a great deal of trouble with the operation and maintenance of coal crushers, because they have to crush not only the coal but such foreign matter as comes through with the coal.

About a year ago we installed on one of the Commonwealth Edison Company's stations, a Bradford Breaker. This breaker contains a large cylinder about 12 ft. in diameter and 20 ft. long in which the coal is slowly rotated. This cylinder is perforated, and the sized coal drops out into a hopper. We find there is no possibility of getting coal that is not properly sized through the breaker. We have used $1\frac{1}{2}$ in. holes for chain-grate stoker practice, and we

believe that $1\frac{1}{4}$ in. may be a little better, as the result of the experience of one year at the Calumet Station with this apparatus.

As to the question of the stratification of the coal in the bunkers; the coal tends to pile up and stick together on the sides of the bunkers, with the result that it will avalanche and successively different sizes will be delivered into the grate. Therefore, it is necessary to exercise the greatest care in the design of bunkers, so as to prevent this action. I have seen coals that would stand almost vertically on the sides of the bunker and under such circumstances it is impossible to overcome the trouble.

As to the question of the distribution of coal to the stoker; my belief is that the best way to handle coal out of the bunker to the grate is with the swinging spout. We have tried out both spouts a number of years, but there will always be better distribution of the coal throughout the grate, if the spout is of the swinging type.

Henry M. Burke In justification of the remark made by Mr. Polakov about not believing in the automatic regulation of stokers, it might be well to mention a practical application which is really a justification of not only Mr. Polakov's remarks but of those who took issue with Mr. Polakov later.

As the operator of a 5,000-hp. industrial plant who has to watch the efficiency from day to day and answer to the treasurer who is spending the money for the fuel, I have found that the automatic control of the total plant is impossible from the standpoint of efficiency and I have also found that the control by manual means is impossible. The plant has been worked on a bonus scheme and we have found that the fireman is anxious to do so much work that when put on absolutely manually handled plan, he is taking care of too many operations to operate this plant most efficiently. We have found also that in operating the plant from the automatic control, that he is too prone to do nothing. Therefore we have worked out standards according to the loads carried by this plant, and have sectionalized the control, taking up the peak loads with manual control and carrying along the regular load by the automatic control.

J. B. Crane. The size of the coal for chain-grate stoker is very important. I have had no experience in burning bituminous coal on chain-grate stokers, but I have had considerable experience in burn-

ing the small sized anthracites, and have found that uniformity of size is essential.

On the question of furnace volume, we started in 1916 to put in boilers with four and a half cubic feet per rated horsepower, and there are very few recent installations that have gone as high as that; but the results are fully warranted. We are finding today, that instead of the melting point of the brick being the deciding factor, it is the temperature at which it begins to compress; and it will begin to compress under a load of 40 to 50 lb. per sq. in. at 1800 to 2000 deg. fahr. Consequently, if combustion chambers are designed so that the load in the hot zone is not more than 25 lb. per sq. in. and so that the combustion chamber is inclined away from the grate, and if some means are provided to overcome the slight tendency for the brick work to fall into the center of the combustion chambers, there should be no trouble with combustion chambers, no matter how heavy loads are carried on the boilers.

Joseph J. Nelis. Mr. Myers has asked why there were no marine stokers. There are two serious limitations in the ship that do not exist in the shore plant.

The sea has not attracted the technical man as yet, and until it does, I do not think we shall get very far with improved machinery.

The second limitation is the space available. A ship is primarily a floating freight. Under the marine classification rules we have certain space allowed for machinery. If we exceed that it damages the registered tonnage and we are penalized by harbor dues, tonnage taxes and other operating costs.

The present marine boiler, in the freight boat particularly, is the Scotch boiler with internal furnaces. It fits the space conditions and is almost fool-proof. This boiler violates all the laws of furnace volume that stationary engineers have been developing through a number of years. A number of very scientific tests made by the Bureau of Mines and the Shipping Board have shown that with practically one-quarter of the volume used in furnaces of stationary boilers, the Scotch boiler gives over 80 per cent efficiency. This boiler has been adapted for marine service because it was found to be the best boiler and it is passing not because it is not the best boiler, but because of increase of steam pressures.

Stokers have been tried on the Scotch boiler, principally under-

feed type. They would burn the coal without difficulty but they could not get rid of the clinkers, so they were finally abandoned.

Some of the large freighters tried chain grates years ago, and they worked fairly well on them. At one time I went round from the east to the west coast on a ship, four of whose boilers were fitted with chain grates. Leaving New York we took the ordinary bunker coal, Pocohantas, and upon arrival at Chili, we took on some Chilian coal, which fused and ran through the grates. We finally threw the chain grates overboard and came into port hand-fired on all boilers.

The marine man has one thing in his favor; that is an absolutely steady load factor.

There are a great number of water-tube boilers afloat now and there are a great many Diesel engines being used. If the stoker manufacturer will coöperate more fully with the marine boiler maker, there is a chance now to put in stokers on ships, as water-tube boilers are being adopted and they have the necessary furnace volume required to make marine stokers successful.

Lester C. Bosler. The coal operator today thinks that the specifications as laid down by the users of the small sizes of anthracite are rather severe. He is called on to produce $\frac{3}{32}$ barley in one case and $\frac{1}{16}$ barley in another, and if the $\frac{3}{32}$ barley is supplied practically all the undersize goes to waste.

We think there should be a reduction in the number of sizes. We would like to elminate the rice and barley as two sizes and combine them into one size, known as "boiler," which would mean a coal through a $\frac{1}{4}$-in. round mesh and over $\frac{1}{16}$-in. round mesh. We would prefer to make this through $\frac{1}{4}$ and over $\frac{1}{32}$-in.

The stoker manufacturers today say this is not an efficient fuel, but we think the time is coming, in order to increase the supply of these sizes, that an endeavor should be made to design a stoker furnace to burn the combination of what is now known as rice and barley.

At the present time we are shipping about 3,000,000 tons of barley every year, and if we could include a little more of the undersize we could probably increase this to nearly 4,000,000 tons, and if we can include the rice in with the new size known as "boiler," the total would be about 8,500,000 tons. All the above will, in turn, increase the amount of business that a stoker of this type could handle.

A. D. White. Previous to five years ago, I had no experience in burning anthracite coals. At the present time, in our plant, there are ten units, three feet wide and 40 feet long, in which anthracite coal is burned successfully. The coal runs anywhere from 38 per cent ash down to 14, and B.t.u. from 7500 up to 11,000. Usually with a shipment of bad coal, we have a hard time until we get rid of it. We run the boilers at about 160 or 180 per cent of rating, burning anthracite rice coal.

R. Sanford Riley. Mr. Myers has asked a few questions and we have tried to answer them. He spoke of fusion on the fuel surface of the underfeed stoker, because of the fuel working upward to the higher temperature zone. All I can say on behalf of the underfeed stoker is that it has no monopoly on the manufacture of clinkers.

Clinkers are the logical result of thoroughly burned-out ash, if the ash has the proper chemical elements. Silica, of course, makes glass and clinkers. Now, as to the troubles encountered in underfeed stokers from these clinkers, which are inherent in certain kinds of coal. I wish to avoid arches because of the reverberatory action under an arch, which raises the temperature, and makes the clinkers still more liquid. Of course manufacturers of chain grates claim that the clinkers which are made do no harm, because they are carried over. That would be ideal if there were no other limitations on the chain-grate stoker. In general the elimination of the arch allows the utilization of the maximum amount of radiation, using radiation as the means of transferring heat from the fuel bed to the boiler. That means cooler fuel bed, less trouble with brick work, and higher efficiency, and of course, less trouble with clinkers.

Mr. Myers also touched on the use of stokers at sea. There are the limitations of space. No doubt we will go, as Mr. Nelis says, to the raising of water tube boilers, because I think that is about the only reasonable possibility. The small combustion chamber in a Scotch boiler is a pretty hard proposition for stokers.

Mr. Myers asked about automatic regulation. I would like to say for one, that we hope that automatic regulation will come in. The difficulty heretofore has been that we have trouble enough to get operators with intelligence enough to regulate the fire directly. Now, when you want them to go one step further, and

have intelligence enough to regulate a regulator, we have not quite come to that. I think the time is coming before long when the regulators will be much more generally used, and that is equivalent to saying that we are getting more intelligent fire-room operators.

Mr. Myers also raised a question regarding setting heights. We did the best we could with the Stokers Manufacturers' Association to establish a rule of thumb that we think good so far as it goes. We do not claim that is perfect, but it is certainly a great advantage to engineers to have some standard established by which some of the most serious mistakes can be avoided.

The point has been made that the stoker is the basis on which boilers should be selected. I believe that is true. The energy is liberated in the furnace. We all agree on that, so that is the starting point in any plant.

W. J. Wohlenberg. Much has been said about automatic control but no very definite idea of the mechanical process that goes on which makes it a definite part of the equipment has been given.

Consider a case of parallel operation, a number of boilers fed, say, by stokers, and having fans operating in parallel. The device which first responds to the load would be one which would change with the change of pressure. This device would operate on a throttle valve, or an electric control which governs, say, the fan. This fan drives the air through the fuel bed.

If a similar change is desired in all parts of the equipment every part of the system must have similar characteristics. That is, all the throttle valves, all the pressure devices and the fans, and even then the right condition will obtain only providing there is the same kind of fuel bed with the same kind of fuel under every boiler. This makes automatic parallel operation very difficult. It seems to me that the people who manufacture automatic control equipment would convey the right impression if instead of calling such equipment automatic control devices they would refer to them as aids in the control of combustion.

H. O. Pond. A stoker might be defined as a mechanical means of introducing fuel into the combustion space. With that thought in mind and to answer the question about burning 75 varieties of fuel, this can be accomplished by pulverizing the fuel.

R. H. Beaumont. When the problem of coal and ash handling comes to us we ask what kind of a stoker is going to be installed, and the owner frequently does not know. The solution of the problem must stop there and await the determination of the type of stoker. Then we make an attempt at solving the problem of properly preparing the coal, of putting it on to the stoker, and of storing, cooling, removing and ultimately disposing of the ashes produced by the stoker. We are handicapped because once having delivered the coal to the stoker, crushed to the proper size, we must take the ashes in the manner in which they are delivered.

The stoker is only one link in the chain and the problem of supplying coal to it and removing ash from it should be solved by coöperation between designers of stokers and coal- and ash-handling equipment.

W. G. Freer said that he was familiar with the Eldon stoker which had been used on locomotives and he felt that it could be adapted to marine boilers. The stoker scattered the coal over the grate.

T. A. Marsh. In all discussions of stoker, or combustion performance, we must keep in mind the ultimate goal — cheap steam production. Sometimes this is accompanied by high boiler and stoker efficiency and sometimes, as in the case where cheap low-grade fuel is burned, by lower efficiency. Investment charges, operating charges, maintenance and ability to burn low-grade cheap coal, are factors in cheap steam costs just as truly as is the efficiency of performance.

Automatic control of air and fuel has received considerable comment in this discussion. From present designs and existing installations it is evident that automatic control of firing operations is making rapid headway. It is the next logical advance step in boiler room operation. No control can be so automatic as to eliminate all manual attention. An ideal situation is automatic control with expert supervision.

Mr. Maynz has inquired regarding the tightness of the compartments of forced-draft traveling grates, — the early troubles of some of such stokers have been practically overcome, so much so that with these stokers, as constructed today, the leakage from compartments is negligible.

Answering Mr. Cary's suggestion as to a different combustion rate unit, I really do not see the purpose of describing the combustion performance by stating the amount of one constituent of the fuel that has been burned, neglecting all others. Sometimes it is much more to the credit of the stoker and furnace that the volatile, rather than the fixed carbon constituent, was consumed.

Actual coal is convenient for figuring capacities of steam output as coal analyses are reported on this basis, also, and not on the basis of fixed carbon.

All the fuel rests on the grate at the beginning of the combustion process. In turn each constituent re-acts in its own manner as subjected to furnace temperatures. Fixed carbon is the last to leave the grate, and its combustion in final analyses is a reaction of gases.

Mr. Cary asked regarding coking coals, — coking coals are unsuitable for chain grates.

The question of anthracite sizes and the modification of existing standards requires a lot of thought and study. Mine equipment for producing, as well as power plant equipment for burning, are both affected.

George I. Bouton. I agree fully with Mr. Cary's statement that it is desirable to eliminate " rousting " and hand-manipulation of overfeed stokers. The way to do this is to provide sufficient draft, and when a plant has out-grown the draft equipment, revise the draft equipment, instead of attempting to substitute a man at the end of a roust bar.

A number of those discussing this group of papers seem to be of the opinion that automatic control is desirable. There is no objection to automatic control where it is properly designed, manufactured, installed and operated. Up to the present time, I have not seen automatic control equipment applied to natural draft stokers which would meet those requirements.

H. F. Lawrence. First, I would like to state that the mechanical stoker is more than a coal feeding mechanism. In addition to putting the coal into the furnace, it must deposit the ashes into some locality from which they can be easily removed, and also furnish the air at such places and velocities and directions to accomplish the burning of the fuel in the best manner.

The underfeed stoker, of course, forms its ash on top. It is the feeding of the coal from the retort that floats the ash on the fuel bed. The coal in addition to rising up, is pushed forward so that the ash is carried forward toward the bridge wall with the fuel.

The underfeed stoker will burn a wide variety of coal. It is built with adjustments, the principal adjustments being the control of the coal from the retorts. That is what makes it possible to burn a wide variety of coals on the underfeed stoker.

We of course must accept the coal as it comes and burn the coal as we get it. Some coals are much worse than others for throwing off cinders. If it is light and friable, as it burns, it does not mat together and that makes more cinders than other coals. Higher setting heights are reducing the cinders.

The use of lower air pressures and velocities as the design is improved will also reduce the quantity of cinders.

High settings are reducing the slag on the tubes. With some fuels this slag is much worse than with others, and provision should always be made in the boiler setting to get at the slag to remove it.

No. 1868

DESIGN OF FLYWHEELS FOR MOTOR-DRIVEN IMPULSE WHEELS

By C. O. Rhys, Atlanta, Ga.
Member of the Society

The author investigates the problem of the design of flywheels for motor-driven impulse machines. He analyzes the characteristics of torque and speed curves of such machines and derives expressions by which the characteristics of these curves may be determined. A method of drawing such curves by means of a transmission dynamometer and steam-engine indicator is described. In appendices the author gives the derivation of the fundamental equations involved and of the working formulas which he presents, and makes calculations for an actual case. He also makes use of the Fourier development of the torque-time diagram as one method of solution.

CONSIDER an impulse machine, such as a punch or shear, which has most of the energy for its working stroke given to a flywheel by the motor during the interval between two strokes. The problem then arises as to what must be the proper size of flywheel to give out this energy during a drop in speed, which will not slow down the motor sufficiently to develop in it an excessive torque.

2 If all the energy used during the working stroke were given out by the flywheel the problem would be a simple one, but this is not the case. As the flywheel drops in speed, the motor does likewise, its torque increasing as its speed decreases. This varying torque integrated with respect to the angle turned through represents an amount of available work in addition to the energy given out by the flywheel.

3 At the conclusion of the working stroke another phase of the problem presents itself. There is a certain time interval before the machine will be required to make another stroke. During this interval the motor acts to accelerate the flywheel, its torque being a maximum at the beginning of the interval, and gradually decreasing as the speed builds up. What will be the speed when the next working stroke is about to be made?

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4 In actual industrial conditions it is very unlikely that there will be an unvarying interval between any two working strokes of such a machine. Consideration of this and the above facts will show that the initial speeds, speed drops, and torques of the different working strokes are by no means identical. It therefore becomes necessary to develop equations which will enable us to investigate the whole range of such possibilities for any combination of flywheel and motor.

5 Starting with the fundamental differential equation for this kind of motion, the author has, during the past two or three years studied various types of solutions for it, checking these with the flywheel and motor performances of a number of commercial machines. From the results of these comparisons he has been able to obtain formulas which are of a simple character and which have uniformly given accurate numerical results.

CHARACTERISTICS OF TORQUE AND SPEED CURVES

6 Before proceeding with the actual development of equations, it may serve to clear the ground if certain general characteristics of the torque and speed curves involved in the problem be indicated. (See Fig. 1.)

7 The resistance-torque curve consists of a series of peaks, representing the working strokes, connected by a (more or less) horizontal line representing the torque necessary to run the machine between strokes. The motor-torque curve is a wavelike line having its minimum value where it intersects the peak at the beginning of a working stroke, and its maximum where it intersects the peak near the end of a working stroke. This is obviously true, since these are the points where the motor-torque curve changes from being greater than the resistance torque curve to being less, and vice versa. From statements previously made it can be seen that these maxima and minima values can vary from peak to peak, depending upon the time dimension between them as indicated in the figure.

8 The height of the resistance curve is obviously the sum of the motor torque and the torque exerted by the flywheel, which latter is represented by the intercept between the two curves.

9 For the range in which we are interested, motor torque is sensibly proportional to motor slip. Slip in this discussion is understood to mean the actual difference between synchronous speed

and the speed under consideration at any instant. Consequently motor torque can be called —

$$Ks$$

where s is the slip and K is a factor or proportionality.

10 The torque exerted by the flywheel during a working stroke is —

$$J \frac{ds}{dt}$$

where J is the mass moment of inertia of the flywheel, and $\frac{ds}{dt}$ is the angular deceleration.

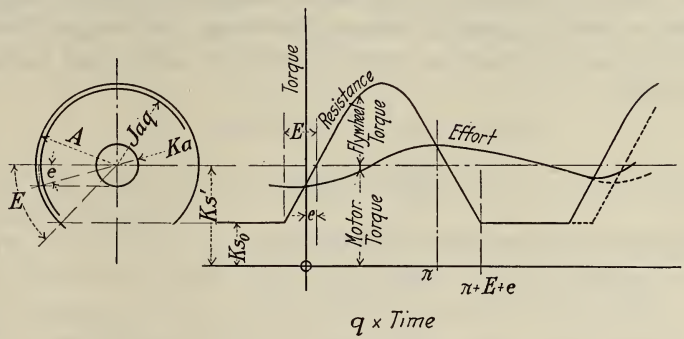

Fig. 1 Torque and Speed Curves

11 From the above follows the differential equation —

$$J \frac{ds}{dt} + Ks = R \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

R, the resistance torque, being some function of the time t. (See Appendix No. 1.)

12 It might be well at this point to transpose Equation [1] to

$$\frac{ds}{dt} = \frac{(R - Ks)}{J}$$

which is zero when $R = Ks$ or the two curves intersect. This agrees with previous statements as to the torque or slip being a maximum or minimum at these points.

13 The solution of Equation [1] is, of course, well known, provided that R is known as a function of t. In general, however, it cannot be derived from known data, and by calculation based on experiment it can only be derived for the particular motor and

flywheel used. However, as will be shown later, motor-torque curves can be taken from a modified steam-engine indicator in conjunction with a transmission dynamometer. These can be developed by means of a Fourier series in such a way that the effect of motor and flywheel changes can be studied. (See Appendix No. 4.)

14 Curves made in this way also bring out the fact that the motor-torque curve very frequently conforms to a sine wave. On this assumption a study of Fig. 1 will show that the motor-torque curve is given by —

$$\text{Motor torque} = Ks' - Ka \cos qt \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where s' is the mean slip during the working stroke, q is the angular velocity of the imaginary vector generating the cosine function, and a is the amplitude of speed fluctuation above and below s'. From this —

$$s = s' - a \cos qt$$

$$\frac{ds}{dt} = aq \sin qt$$

and Equation [1] can now be written as —

$$Jaq \sin qt + Ks' - Ka \cos qt = R$$

or

$$Jaq \sin qt + Ks' - Ka \cos qt = Ks' + A \sin (qt - e) \quad . \quad [3]$$

where $A = a(J^2q^2 + K^2)^{\frac{1}{2}}$ and $\tan e = \dfrac{K}{Jq}$.

15 R and s are each expressed as a function of the time by two terms of a Fourier development. The R and s curves have a phase difference of $\dfrac{\pi}{2} - e$. From Equation [3] formulas can be obtained (see Appendix No. 2), from which all necessary numerical results can be computed.

16 The above solution of the differential Equation [1] applies within the limits of the working stroke, that is to say, while the flywheel is decreasing in speed. We have now to consider what happens in the interval between two working strokes when the flywheel speed is increasing, and the resisting torque can be considered as constant, viz., the torque necessary for running the machine light. For these conditions,

$$-J\frac{ds}{dt} + Ks = Ks_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

s_0 being the slip at which the motor would run the machine light indefinitely. The solution of [4] is —

$$t_a = \frac{J}{K} \log_e \left(\frac{s_1 - s_0}{s_2 - s_0}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

or

$$s_2 = e^{-\frac{K}{J}t_a} (s_1 - s_0) + s_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

t_a is the time between working strokes, s_1 the slip at the end of a working stroke, and s_2 the slip at the beginning of the next one.

17 From solutions [3], [5] and [6] Table 1 of formulas for practical calculation has been worked out. (See Appendix No. 2.) All these formulas are quite simple in character and provide means for estimating what may be expected to occur during a working stroke, whatever time may have elapsed since the preceding one.

HOW THE FORMULAS ARE APPLIED

18 We can now show how the formulas can be applied in practical calculation.

19 In any case we must know, to start with, the amount of work which the machine does in one stroke. If the machine exists only on paper, and no machine doing similar work is available, this must be estimated in the most convenient way. If, however, an existing machine be available for testing purposes, the amount of energy required per stroke can be found by a method described by the writer in *Mechanical Engineering* for July 1921.

20 From the product of this amount of energy and the number of working strokes per minute (plus a percentage depending upon the nature of the problem) the horsepower of the required motor can readily be determined. Its synchronous speed will depend upon the gear ratio and other design factors, into which it is unnecessary to enter here. At this point we have sufficient information to determine the quantities K, W, and θ in Table 1.

21 As a first approximation, at any rate, w', the mean speed during the stroke, can be taken as the full-load speed. This fixes s', the mean slip, and also permits the computation of the quantity I (the torque-time integral or angular impulse) and t_w (the time for a working stroke).

22 We now come to the quantity E. A glance at the diagram, Fig. 1, will show the meaning of this quantity better than it can be explained in words, but it might be described as the angle turned

through by the imaginary vector generating the resistance-torque curve while the curve is passing from the ordinate value Ks_0 to Ks', that is, from light running resistance torque to resistance torque at mean slip.

23 This quantity E can now be computed by the short formula

TABLE 1 FLYWHEEL AND MOTOR FORMULAS

Symbol	Description	Unit	Formula
A	Amplitude of resistance-torque curve	Pound-feet	$\frac{K(s'-s_0)}{\sin E}$
a	Amplitude of speed fluctuation	Radians per sec.	$\frac{A}{\sqrt{J^2q^2+K^2}}$
[1]E	Angular lag of A-curve from $s = s_0$	Radians	$\sqrt{\frac{Kt_w(s'-s_0)}{I-Ks't_w}+\frac{\pi^2}{16}}-\frac{\pi}{4}$
	Angular advance of motor-torque curve over A-curve	Radians	$\tan e = \frac{K}{JQ}$
I	$\int Tdt$ for motor shaft during stroke	Ft-lb-sec.^{2}	$\frac{Wt_w}{\theta}$ or $\frac{W}{\omega'}$
J	Mass moment of inertia of flywheel on motor shaft	Ft-lb-sec.^{2}	$\frac{1}{q}\sqrt{\left(\frac{A}{a}+K\right)\left(\frac{A}{a}-K\right)}$
K	Factor of proportionality of slip to motor torque	Ft-lb-sec.	$\frac{50,150 \times \text{hp.}}{(\text{Synchr. r.p.m.} - \text{F.L. r.p.m.}) \times \text{F.L. r.p.m.}}$
q	Angular velocity of radii of circles generating curves	Radians per sec.	$\frac{\pi+2E}{t_w}$
s'	Mean slip during stroke	Rad. per sec.	$s_2 + a$
s_0	Light running slip	Rad. per sec.	.....
s_1	Slip at end of stroke	Rad. per sec.	$s' + a\cos(E+e)$
s_2	Slip at beginning of stroke	Rad. per sec.	$-\frac{Kta}{\epsilon J}(s_1-s_0)+s_0$ [or $s' - a\cos(E-e)$]
T	Torque at any instant	Pound-feet	$T = Ks$
t_a	Time for flywheel to regain speed	Seconds	$\frac{J}{K}\log_e\left(\frac{s_1-s_0}{s_2-s_0}\right)$
t_w	Time for working stroke	Seconds	$\frac{\theta}{\omega'}$ or $\frac{60}{\text{strokes per min.}}$
W	Work per stroke	Foot-pounds	.....
θ	Angular displacement of motor shaft during stroke	Radians	.....
ω'	Mean speed during stroke	Rad. per sec.	.....

[1] $F(E) = (\pi + 2E)\tan E - \frac{2Kt_w(s'-s_0)}{I-Ks't_w} = 0$ { Use if $E > 15^\circ$ } $F'(E) = 2\tan E + \frac{\pi+2E}{\cos^2 E}$

given in the table. If most of the work during the working stroke be done by the flywheel, as is generally the case, E will come out less than 15 deg. and be quite accurate enough. It sometimes happens, however, in the case of small peak loads of frequent occur-

rence, that a powerful motor is used in conjunction with a light flywheel, when E may be considerably greater than 15 deg. In these cases E must be computed by the longer method shown in Appendix No. 2.

24 Knowing E, we can compute A, and then have the resistance torque as a function of the time, expressed by the first two terms of a Fourier development. It might be stated here that the great majority of impulse machines, whose work consists of punching, shearing, applying pressure, etc., actually show resistance curves whose shape can be closely approximated by a function of this general character. In cases where this is not sufficiently close, it becomes necessary either to use more terms of a Fourier development or to proceed by the "short arc" method, as described in Appendix No. 4. This involves more work, especially if the stroke is to be studied over any considerable range, and it is not often necessary as the drop in speed of a flywheel depends much more upon the diagram's area than upon its shape.

25 The quantity a, or the amplitude of speed fluctuation, can be assumed according to the number of working strokes per minute and the allowable momentary overload of the motor. This completes all the values necessary for the calculation of J, the mass moment of inertia of the flywheel.

26 For the first stroke a can be taken as $s' - s_0$, in which case E and e are identical. For the second stroke we can assume a slightly greater value for s' and therefore for w', and compute t_w, I, E, q, e, and A to correspond. J, of course, is the same as before, so we must calculate another a from the table; s_2 must now be computed as the slip at the beginning of the second stroke. We can now compute t_a, using this value of s_2 and of course the s_1 of the first stroke.

27 Assuming still greater values of s' for successive strokes, and proceeding in like manner, we can compute a table as shown in Appendix No. 3, giving speed drops, torques, etc., for strokes made with all kinds of time intervals between them. In this way we can investigate, as fully as desirable, all the conditions under which the motor may have to perform.

APPENDIX NO. 1

DERIVATION OF FUNDAMENTAL EQUATIONS

28 Consider a flywheel and motor set. This can be regarded as a conservative system having a potential function. When kinetic energy is taken from the flywheel by the periodic load, the drop in speed causes an increased torque in the motor. When the load period comes to an end, the motor torque is usually just past its maximum value, and the work which it does in falling back to the torque at the commencement of the next stroke, is employed in increasing the speed of the rotating parts and thus is converted into kinetic energy. In other words, the kinetic energy lost by the flywheel during a stroke can be regarded as though it were stored up in the motor in the form of potential energy. This is really analogous to the transformation of energy which is continually taking place from kinetic to potential and vice versa in such a device as a pendulum. The only difference is that the "potential" function is electrical instead of being gravitational.

29 Consider Lagrange's equation for motion of this kind, with the usual notation —

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}}\right) - \frac{\partial L}{\partial q} = 0$$

where L is kinetic energy – potential energy.

30 Let T = motor torque. The "potential" energy is then $\int (R - T)\, d\theta$, that is, the energy given out by the flywheel, and, according to the above conception, "stored up" in the motor. The kinetic energy is $\frac{1}{2}Jw^2$. θ corresponds to q and w to $\dot{q}$.

Then —

$$L = \tfrac{1}{2}Jw^2 - \int (R - T)\, d\theta$$

$$\frac{\partial L}{\partial \dot{q}} \text{ or } \frac{\partial L}{\partial w} = Jw \text{ and } \frac{d}{dt}(Jw) = J\frac{dw}{dt}$$

Since $w = w_1 - s$, where w_1 = synchronous speed and s = slip,

$$J\frac{dw}{dt} = \frac{d}{dt}(Jw_1 - Js) = -J\frac{ds}{dt}$$

Also —

$$\frac{\partial L}{\partial q} \text{ or } \frac{\partial L}{\partial \theta} = -(R - T)$$

so that Lagrange's equation becomes —

$$J\frac{ds}{dt} + T = R$$

or, if torque be proportional to slip for the range under consideration, $T = Ks$ and we get —

$$J\frac{ds}{dt} + Ks = R$$

This is the differential equation for the working stroke.

31 For the period of acceleration between strokes the "potential" function is —

$$\int (T - R)\, d\theta$$

and the kinetic energy is the same as before or $\frac{1}{2}Jw^2$, so that L becomes —

$$\tfrac{1}{2}Jw^2 - \int (T - R)\, d\theta$$

and Lagrange's equation reduces to —

$$-J\frac{ds}{dt} + T - R = 0$$

for the period between strokes. The quantity q in the above, which is a generalized coördinate of Lagrange's equation, has nothing to do with the q representing radians per second which appears elsewhere.

APPENDIX NO. 2

DERIVATION OF WORKING FORMULAS GIVEN IN TABLE 1

32 The diagram of Fig. 1 is a graphic representation of Equation [2] in Par. 14. Vertical ordinates represent torque and horizontal ordinates represent time multiplied by the quantity q, which is the (constant) angular velocity of the imaginary vectors generating the sine curves. The zero is chosen at the point where the motor-torque curve at its minimum cuts the resistance-torque curve, and this gives the intersection of the two curves for maximum motor torque at the ordinate corresponding to $qt = \pi$. E is the angular displacement from the point where the resistance torque begins to the ordinate of average height of the motor-torque curve, and e is the phase difference between the motor-torque curve and the resistance-torque curve, less 90 deg.

33 The angular impulse (or $\int R dt$ integral) for the motor shaft during a working stroke is given by —

$$\int [A \sin(qt - E) + Ks']\, dt$$

with the proper limits inserted. With the zero point as above these limits are from $(\pi + 2E)/q$ to 0, so that we have —

$$\text{Angular impulse} = I = \frac{1}{q}\left[2A \cos E + Ks'(\pi + 2E)\right] \quad . \quad . \quad . \quad . \quad . \quad [7]$$

34 If the time for the whole working stroke be t_w, then $qt_w = \pi + 2E$ and —

$$q = \frac{(\pi + 2E)}{t_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

35 From [7] and [8],

$$I(\pi + 2E) = t_w[2A \cos E + Ks'(\pi + 2E)]$$

and —

$$A = \frac{(I - Ks't_w)\ (\pi + 2E)}{2t_w \cos E} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

36 From the diagram,

$$A = \frac{K(s' - s_0)}{\sin E} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

and from [9] and [10],

$$(\pi + 2E) \tan E = \frac{2t_w K(s' - s_0)}{I - Ks't_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

37 If E be small, so that $\tan E$ is approximately equal to E, [11] reduces to —

$$E = \sqrt{\frac{K(s' - s_o)t_w}{I - Ks't_w} + 0.615} - 0.7854$$

but if E comes out with a value greater than about 15 deg., then Equation [11] must be solved as it stands. This can be done most readily by writing —

$$f(E) = (\pi + 2E) \tan E - \frac{2Kt_w(s' - s_0)}{I - Ks't_w} = 0 \quad . \quad . \quad . \quad . \quad [12]$$

$$f'(E) = 2 \tan E + \frac{\pi + 2E}{\cos^2 E}$$

then if a trial value of E, say E', be taken,

$$E' - \frac{f(E')}{f'(E')}$$

will be a closer approximation to that value of E, which satisfies Equation [12]. The process can of course be carried on to obtain any desired degree of accuracy.

38 From Equation [3], Par. 14, A is seen to be $a\sqrt{J^2q^2 + K^2}$, from which

$$J = \frac{1}{q}\sqrt{\left(\frac{A}{a} + K\right)\left(\frac{A}{a} - K\right)}$$

and s_1 the slip at the end of a working stroke is very evidently $s' - a \times \cos(\pi + E + e)$, or $s' + a \cos(E + e)$.

APPENDIX NO. 3

CALCULATION OF A PRACTICAL CASE

39 Consider a machine which absorbs 2000 ft-lb. of energy per stroke and which may operate often enough to require a 2-hp. motor. Synchronous speed, 1800 r.p.m.; full-load speed, 1730 r.p.m.

$$K = \frac{50{,}150 \times 2}{70 \times 1{,}730} = 0.83$$

40 Suppose the gear ratio from motor shaft to operating shaft to be 10 to 1, and the working stroke to be done in one revolution of the operating shaft; then —

$\theta = 62$ radians

$$w' = \frac{1{,}730 \times 3.14}{30} = 181 \text{ radians per sec.}$$

$$s' = \frac{(70 \times 3.14)}{30} = 7.3 \text{ radians per sec.}$$

Assume —

$s_0 = 2$ radians per sec.

$$t_w = \frac{62}{181} = 0.343 \text{ sec.}$$

$$I = \frac{2{,}000}{181} = 11.05 \text{ ft-lb-sec.}$$

$$E = \sqrt{\frac{0.83 \times 5.3 \times 0.343}{11.05 - (0.83 \times 0.343 \times 7.3)} + 0.615 - 0.7854} = 0.1 \text{ radian or } 5.73 \text{ deg.}$$

The value of E is small enough to be within the limits of accuracy of this formula.

$$A = \frac{(0.83 \times 5.3)}{\sin 5.73^0} = 44 \text{ lb-ft.}$$

$$q = \frac{3.14 + 0.2}{0.343} = 9.72 \text{ radians per sec.}$$

If we now consider the amplitude a to be 5.3, that is, for the flywheel to be at light running speed when the working stroke begins, then $\frac{A}{a} = \frac{44}{5.3} = 8.3$ and J, the mass moment of inertia of the flywheel,[1] is —

$$J = \frac{\sqrt{(8.3 + 0.83)(8.3 - 0.83)}}{9.72} = \frac{0.85 \text{ ft}^2\text{-lb.}}{32.2}$$

41 If we now take s' for the second stroke as 10 radians per sec., we find that $t_w = 0.348$ sec., $I = 11.25$ ft-lb-sec., $E = 0.16$ radian or 9.16 deg., $q = 9.95$ radians per sec., and $A = 41.5$ lb-ft. J is still 0.85 and therefore a becomes —

$$\frac{41.5}{\sqrt{(0.85 \times 9.95)^2 + 0.83^2}} = 4.88 \text{ radians per sec.}$$

e for this stroke is —

$$\tan^{-1}\left[\frac{0.83}{0.85 \times 9.95}\right] = 0.098 \text{ radian}$$

and $E - e = (0.16 - 0.098) = 0.062$, so that s_2 the slip at the beginning of this stroke is —

$$10 - [4.88 \times \cos (0.062 \text{ radian})] = 5.12 \text{ radians per sec.}$$

and s_1, the slip at the end of the *first* working stroke is found to be $8.3 + [5.3 \cos (0.2 \text{ radian})]$ or 13.5 radian per sec., so that t_a, the time interval between the strokes, is —

$$t_a = \frac{0.85}{0.83} \times \log_e \left(\frac{13.5 - 2}{5.12 - 2}\right) = 1.34 \text{ sec.}$$

[1] This is the mass moment of inertia of a flywheel on the motor shaft. For the same flywheel effect on any other shaft, the mass moment of inertia varies as the square of the speed ratio between the shafts.

42 Proceeding in this way we can tabulate results as below:

Stroke No.	A	a	E	e	I	q	s'	s_1	s_2	t_a	t_w
1	44.0	5.3	0.10	0.10	11.05	9.72	7.3	13.5	2		0.343
										1.34	
2	41.5	4.91	0.16	0.098	11.25	9.95	10.0	14.75	5.1		0.348
										0.88	
3	40.0	4.65	0.207	0.091	11.35	10.10	12.0	16.46	7.4		0.352

which will enable us to study the effect of working strokes beginning at any speed of the motor, or after any time interval between strokes.

APPENDIX NO. 4

DEVELOPMENT OF THE DIFFERENTIAL EQUATION $\frac{Jds}{dt} + Ks = R$ FOR THE GENERAL CASE WHERE $R = f(t)$

43 Rearranging the above equation, we have —

$$\frac{ds}{dt} + \frac{Ks}{J} = \frac{R}{J} \quad \ldots\ldots\ldots\ldots \quad [13]$$

44 R is to be developed as a Fourier series in the form —

$$A_0 + A_1 \sin qt + B_1 \cos qt + A_2 \sin 2qt + B_2 \cos 2qt \ldots\ldots$$

or, more briefly,

$$f(t) = A_0 + \sum_{n=1}^{n=\infty} A_n \sin nqt + B_n \cos nqt$$

45 A_0 is the average height of the curve, or the average torque, or K multiplied by the average slip $= Ks_a$. Equation [13] now becomes —

$$\frac{ds}{dt} + \frac{Ks}{J} = \frac{Ks_a}{J} + \frac{1}{J}\sum_{n=1}^{n=\infty} A_n \sin nqt + B_n \cos nqt$$

the solution of which is —

$$s = \epsilon^{-\frac{K}{J}t}\left\{\int\left(\frac{Ks_a}{J}\,\epsilon^{\frac{K}{J}t} + \frac{\epsilon^{\frac{K}{J}t}}{J}\sum_{1}^{\infty} A_n \sin nqt + B_n \cos nqt\right)dt + C\right\}$$

Integrating,

$$s = \epsilon^{-\frac{K}{J}t}\left\{ s_a \epsilon^{\frac{Kt}{J}} + \frac{\epsilon^{\frac{K}{J}t}}{J}\sum \frac{A_n\left(\frac{K}{J}\sin nqt - nq\cos nqt\right) + Bn\left(\frac{K}{J}\cos nqt + nq\sin nqt\right)}{\left(\frac{K^2}{J^2} + n^2q^2\right)} + C\right\}$$

Now $$\frac{K}{J}\sin nqt - nq\cos nqt = \sqrt{\frac{K^2}{J^2} + n^2q^2}\,\sin(nqt - e_n)$$

and $$\frac{K}{J}\cos nqt + nq\sin nqt = \sqrt{\frac{K^2}{J^2} + n^2q^2}\,\sin(nqt + E_n)$$

where $\cot e_n = \tan E_n = \dfrac{K}{Jqn}$ and $e^{\frac{K}{J}t}\ -\frac{K}{J}t = 1$, so that the above equation simplifies to —

$$s = s_a + \sum_1^\infty \frac{A_n \sin(nqt - e_n) + B_n \sin(nqt + E_n)}{\sqrt{K^2 + J^2 n^2 q^2}} + C\epsilon^{-\frac{K}{J}t} \quad . \; . \; . \; [13a]$$

46 Let $s = s_0$ when $t = 0$ at the beginning of a working period, and we get —

$$s_0 = s_a + \sum_1^\infty \frac{B_n \sin E_n - A_n \sin e_n}{\sqrt{K^2 + J^2 n^2 q^2}} + C^*$$

where $C^* \epsilon^{-\frac{K}{J}\cdot 0} = 1$.

Finding C from this and inserting this value in Equation [13a], we have —

$$s - s_0 = (s_a - s_0)\left(1 - \epsilon^{-\frac{K}{J}}\right) - \epsilon^{-\frac{K}{J}t}\sum_1^\infty \frac{B_n \sin E_n - A_n \sin e_n}{\sqrt{K^2 + J^2 n^2 q^2}}$$

$$+ \sum_1^\infty \frac{A_n \sin(nqt - e_n) + B_n \sin(nqt + E_n)}{\sqrt{K^2 + J^2 n^2 q^2}} \quad . \; . \; . \; . \; . \; . \; [14]$$

47 This is the general solution for $s - s_0$, or the drop in speed at any time t below the speed at the commencement of the working period. To calculate the actual values we must know all the coefficients A_n and B_n of the Fourier series.

48 When the motor has been running for some time, or t has become large, $\epsilon^{-\frac{Kt}{J}}$ approaches zero, and Equation [14] above approaches the form —

$$s = s_a + \sum_1^\infty \frac{A_n \sin(nqt - e_n) + B_n \sin(nqt + E_n)}{\sqrt{K^2 + J^2 n^2 q^2}} \quad . \; . \; . \; . \; . \; [15]$$

49 This transformation is due to the fact that, as time goes on, the terms containing $\epsilon^{-\frac{K}{J}t}$ tend to "damp" themselves out of existence. In most practical cases $\epsilon^{-\frac{K}{J}t}$ would be negligible at the end of a very few minutes' running of a machine.

50 It might be noted here that if it be accurate enough to express the resistance torque by a simple sine function of the time, $A \sin qt$, then [**15**] reduces to —

$$s = s_a + \frac{A_1 \sin (qt - e_1)}{\sqrt{K^2 + J^2q^2}}$$

$$s_{max} = s_a + \frac{A_1}{\sqrt{K^2 + J^2q^2}}$$

and, by giving e the proper value, s would become $s_a - a \cos qt$, a being $\frac{A}{\sqrt{K^2 + J_1^2q^2}}$ corresponding with the formulas given in the working table.

51 To get the coefficients A_n and B_n suppose we have a diagram showing torque and time. The author has obtained many of these by means of a transmission dynamometer fitted with an ordinary steam-engine indicator, whose pencil marks on a strip of paper moving at a constant speed. Such an arrangement is shown in Fig. 2. The driven machine has some temporary motor and flywheel which will make it run. We therefore know J, and the periodic time t_1 can be found from the diagram. $q = \frac{2\pi}{t_1}$; K is a characteristic of the motor, while e_n for each term can be found from J, q, n and K. s_a is the average height of the diagram divided by K, and from the properties of a Fourier series,

$$A_n = \frac{2\sqrt{K^2 + J^2n^2q^2}}{t_1} \int_o^{t_1} s \sin (nqt - e_n)dt$$

$$B_n = \frac{2\sqrt{K^2 + J^2n^2q^2}}{t_1} \int_o^{t_1} s \cos (nqt - e_n)dt$$

s being of course the torque anywhere divided by K. The constants A_n and B_n can now be developed by the application of the usual analytic and graphic methods, the graphic method being specially recommended for the development of arbitrary functions. It is now possible to plot the slip curves of other combinations of flywheel and motor by substituting the proper values of J and K.

52 There remains another method of studying motor and flywheel effects on a machine. By driving the strip of paper past the indicator pencil from some convenient shaft of the dynamometer, we can get a diagram giving torque and angular displacement. The equation —

$$J \frac{ds}{dt} + Ks = R$$

can then be written —

$$Jw \frac{ds}{d\theta} + Ks = R$$

R being resistance torque and w angular velocity. J must be assumed, s, as before, is torque divided by K, and $w = w' - s$, where w' is initial angular velocity.

53 To plot the R-curve it is necessary to know $\frac{ds}{d\theta}$, to do which there is a choice of the following methods:

1 Draw many equidistant ordinates and divide the change in torque by the change in angle and by K

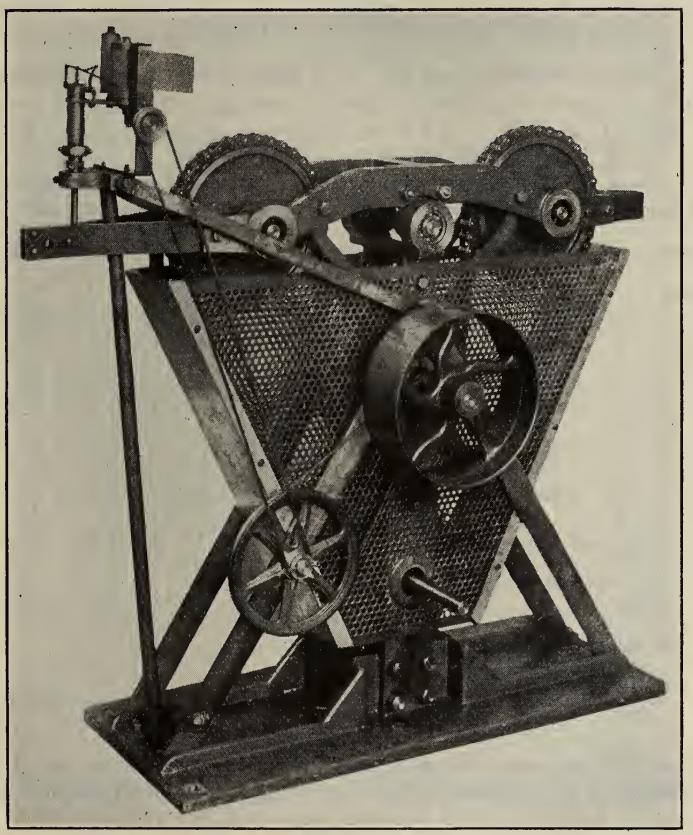

Fig. 2 Transmission Dynamometer Fitted with a Steam-Engine Indicator for Drawing Torque and Angular Displacement Curves

2 Develop the curve in the form of a Fourier series, $Ks = A_o + A_1 \sin p\theta + B_1 \cos p\theta + A_2 \sin 2p\theta$, etc., from which

$$\frac{ds}{d\theta} = \frac{1}{K}\left[A_1 p \cos p\theta - B_1 p \sin p\theta, \text{ etc. } . .\right]$$

3 Use a mechanical differentiator.

54 In any case, supposing the R-curve to have been obtained, we can now solve the equation —

$$\frac{ds}{d\theta} = \frac{R - T}{Jw}$$

and proceed as follows:

55 Let $\delta\theta$ be a small increment of angular displacement, the smaller the more accurate. We know or can assume the value s of the slip at the beginning of the working stroke, and also R and w. A trial value of J can be assumed. Then $\frac{ds}{d\theta}$ can be calculated and also δs, a small increment in the torque, since $\delta s = \delta\theta\left(\frac{ds}{d\theta}\right)$. $w = w' - s$, and T, the effort torque, can be taken as Ks, or can be read from the torque-speed performance chart of the proposed motor. The small portion of a table below will show the method more easily than it can be explained. Take J as $0.5\ \frac{\text{ft-lb.}}{32.2}$ Other units are in feet, pounds, radians and seconds.

$\delta\theta$	$ds/d\theta$	δs	R	T	s	w
	0.016		3	2	1.0	125
1		0.016				
	0.096		10	4	1.02	125
1		0.096				
	0.206		17	4.1	1.11	125
1		0.206				
	0.380		28	4.4	1.32	124
1		0.380				
	0.576		41	5.4	1.70	124

56 Starting with $R = 3$, $T = 2$, $w = 125$, $s = 1$,

$$\frac{ds}{d\theta} = \frac{3 - 2}{0.5 \times 125}$$

or 0.016. The increment $\delta\theta$ having been taken as 1 for simplicity, $\delta s = 1 \times 0.016$ or 0.016. $w = 125 - 0.016$ or almost no change. The new value of s is 1 plus the increment 0.016 or 1.02 in round figures, which corresponds to a torque, taken from a motor torque-speed chart, of 4 lb-ft. The next value of $\frac{ds}{d\theta}$ is $\frac{10 - 4}{0.5 \times 125}$ or 0.096. The process can now be continued through one or more complete cycles of the machine's action.

DISCUSSION

Alphonse A. Adler asked if the author's formula could be used for conditions involving direct-current motor drive. Also what

modifications, if any, could be made in the relation of torque to drop in speed in direct-current motor drives so that Lagrange's equation might be applicable.

THE AUTHOR. The formula in question can be used for any form of motor provided that the torque varies directly as the slip, in other words, as long as the relation

$$T = Ks$$

holds. If the torque is some other function of the slip or, in fact

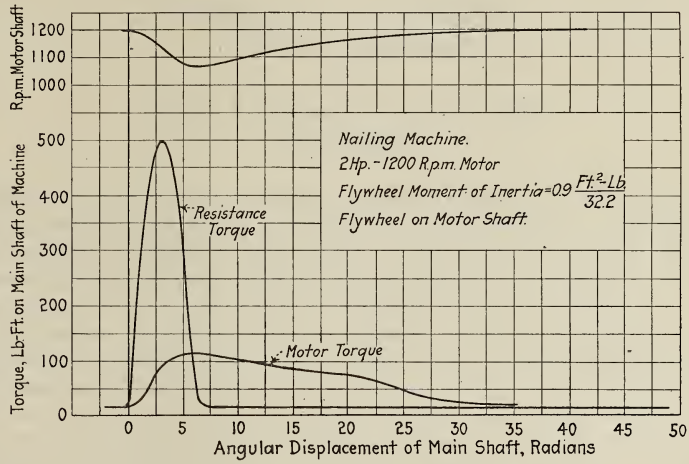

FIG. 3 TORQUE-SPEED CHART OF MOTOR

any function of the slip, then Lagrange's equation, as shown in Par. 30, becomes,

$$J.ds/dt + T = R$$

T and R being *any* functions of the time. This can be thrown into the form

$$J.w.ds/dO + f(s) = F(O)$$

I have not been able to discover a general solution of this differential equation, which can be expressed in terms of ordinary functions. It can, however, be developed by the short-arc method shown in Par. 55, the values of $F(O)$ being known from the characteristics of the driven machine, and the values of $f(s)$ being read from the torque-speed chart of the proposed motor. I have plotted a large number of curves for different motors and machines in this way, one of which is shown in Fig. 3.

No. 1869

EFFECT OF PULSATIONS ON FLOW OF GASES

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Non-Member

The movement of gases and liquids when calculated by the existing hydraulic formulas presupposes a steady or continuous flow of the fluid. Anything which causes this flow to proceed in puffs, waves, or pulsations will result, by the action of metering devices, in errors often of great magnitude which generally do not admit of any adjustment, or of any definite knowledge of the amount of the error.

The present paper discusses work undertaken under the joint direction of the Engineering Experiment Station of the Ohio State University and the Research Sub-Committee on Fluid Meters of The American Society of Mechanical Engineers, which had for its object (a) *the study of the nature of the pulsation and* (b) *the discovery of some practical means of reducing or eliminating the pulsation or of compensating for its effects on the devices used for measuring fluid flow. The investigation was confined to the venturi meter, the orifice meter, the flange nozzle meter, and the pitot meter, using air flow from a small compressor discharging into a 3-in. line. It is believed, however, that the basic principles established by the experiments are fundamental for pulsating-flow conditions for gas, steam, and water as well as for air, and also for other sizes and kinds of installations.*

ONE of the most disturbing factors encountered in recent years in the metering of air, gas, steam, and water, especially in connection with all forms of power engineering, has been that due to turbulent or pulsating flow. This has not been confined to any one class or type of meter, but is present to a more or less degree with all forms of metering devices.

2 The measurement of gases and liquids when calculated by the existing hydraulic formulas presupposes a steady or continuous

[1] Robinson Research Fellow, The Ohio State University.

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

flow of the fluid. Anything which causes this flow to proceed in puffs, waves, or pulsations will result, by the action of metering devices, in errors often of great magnitude which generally do not admit of any adjustment, or of any definite knowledge of the amount of the error.

3 The work described in the present paper was undertaken under the joint direction of the Engineering Experiment Station of The Ohio State University, and the Research Sub-Committee on Fluid Meters of The American Society of Mechanical Engineers. A sub-committee of the Fluid Meters Committee consisting of A. R. Dodge, H. N. Packard, and H. Judd was selected to take direct charge of the research work.

PURPOSE OF THE INVESTIGATION

4 The object of the investigation as outlined by the sub-committee in direct charge was twofold:

a To study the nature of the pulsation

b To discover some practical means of reducing or eliminating the pulsation, or of compensating for its effects on the devices used for measuring fluid flow.

5 As the work has proceeded it has been necessary to use a specific installation involving the flow of air only, and also to limit somewhat the scope of the investigation. The work is by no means considered to be complete, and it is the intention in the future to continue with the research so as to shed more light on many doubtful points that have arisen, as well as to try out a number of new suggestions.

6 For convenience, flow meters have been classified in two main divisions which may be called (1) positive meters, and (2) inferential meters. The domestic gas, or water, meter is an example of the first class. It is a displacement meter in which an actual volume of gas is introduced into a container of known size, and the quantity thus measured is registered on the meter.

7 In commercial installations of even moderate size, inferential meters are used almost entirely. In meters of this class some function of the quantity of fluid passing a given cross-section of pipe is measured and from this observation the actual flow is deduced or "inferred." Perhaps the most common function observed

in such meters is a pressure difference which is a quadratic function of the velocity of flow. This method can be made to give accurate results under steady-flow conditions, but when the flow is pulsating the accuracy of the measurement is seriously affected, if, indeed, not entirely destroyed.

8 Three general cases may be mentioned where this problem is of great importance: (1) The measurement of natural gas, both entering and leaving a compressor station where reciprocating compressors are used; (2) the measurement of air both entering and leaving large reciprocating air compressors, or blowing engines; and (3) the measurement of steam supplied to reciprocating steam engines. The steam flow is pulsating in character because the engine cuts off the steam supply during a considerable part of each stroke. In each of these cases the flow of the fluid has a regular, comparatively rapid, rhythmical pulsation, which occasions serious errors in measurement, especially where the measuring element is of the inferential type.

9 Similar pulsating conditions are present in water flow where reciprocating pumps are used. The problem, however, is more easily solved by the proper use of air chambers and surge tanks. Water hammer in pipe lines from whatever cause bears a striking similarity to the pulsating effect produced by an air-compressor valve.

10 We have confined our investigations to inferential meters. These meter elements as selected are the venturi meter, the orifice meter, the flange nozzle meter, and the pitot meter. Furthermore, we have been limited to air flow from a small compressor discharging into a 3-in. line; and hence our findings, strictly speaking, would be applicable only to installations of similar character. However, it would seem highly probable that the basic principles established by these experiments would be fundamental for pulsating-flow conditions for gas, steam, and water as well as for air, and also for other sizes and kinds of installations.

EQUIPMENT EMPLOYED IN THE INVESTIGATION

11 The experimental work was carried on in the Mechanical Engineering Laboratories of the Ohio State University, and was begun in May, 1920. A large part of the work was done during the summer vacations of 1920–1921. During the remainder of the two years devoted to the work such time was put in as could be spared

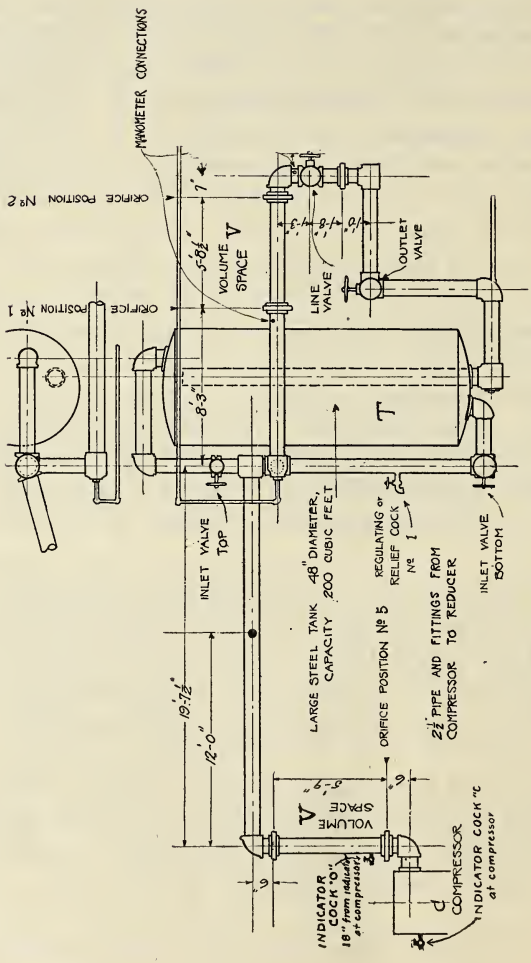

FIG. 1 GENERAL LAYOUT OF APPARATUS — LEFT-HAND PORTION

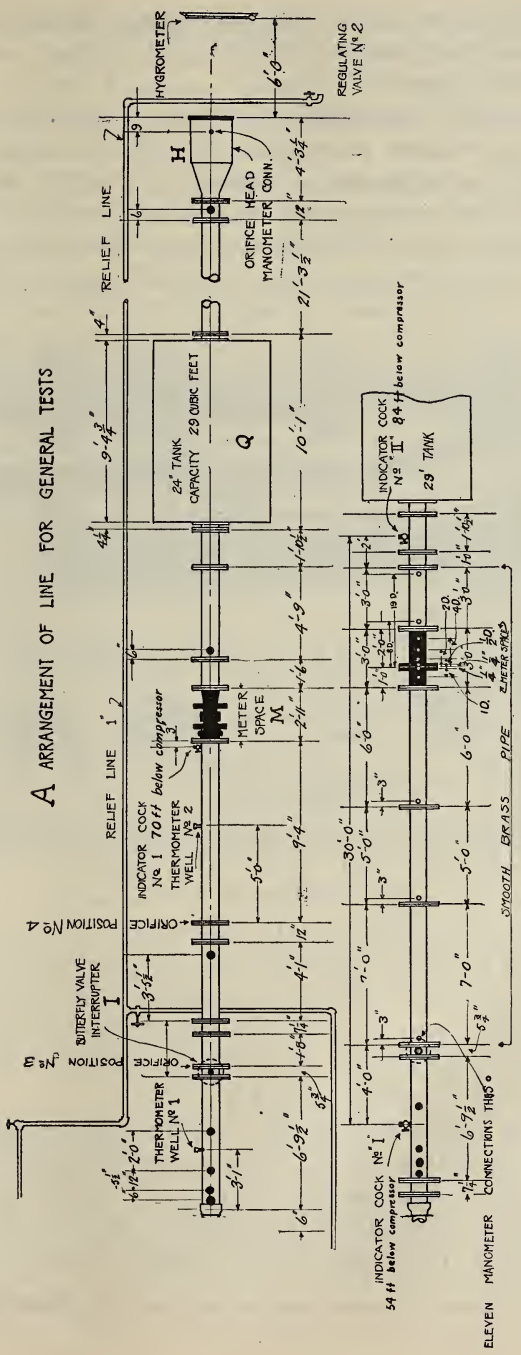

FIG. 1 GENERAL LAYOUT OF APPARATUS, RIGHT-HAND PORTION

from the university schedule. Acknowledgment should be made of the valuable service rendered by Mr. Paul Bucher,[1] especially during the vacation periods.

12 The essential elements for carrying on our project were: (1) a disturbing element to produce the pulsating flow; (2) a quieting element, or elements, to eliminate or modify the pulsations; and (3) a measuring element, to indicate constant flow conditions and also to indicate the effect of the pulsating flow.

13 *Disturbing Elements.* Fig. 1 shows the general layout of the apparatus, at the extreme left hand of which is located the air

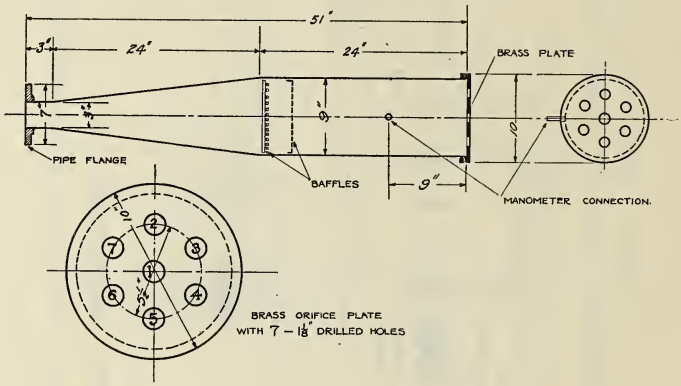

Fig. 2 Orifice Head

compressor *C*, a 9-in. by 9-in. single-stage, single-acting, gas engine-driven machine running at 293 r.p.m. This supplied air to a line about 120 ft. in total length of which 50 ft. was made up of 2½-in. pipe containing several short lengths and fittings. The remaining 70 ft. comprised the 3-in. test line of straight continuous length. This test line was at first made up of standard 3-in. black pipe of commercial quality; later 24 ft. of 3-in. brass pipe was substituted for that portion of the test line preceding the meter and extending 3 ft. below the meter section. (See Fig. 1, *B*.)

14 This air supply with its full pulsating effect could be admitted directly to the test line or could be first discharged through a large tank before entering the test line. A second disturbing element for producing pulsations artificially is shown at *I*, Fig. 1, the butterfly-valve interrupter. This butterfly valve could be

[1] Assistant Professor of Steam Engineering, The Ohio State University.

driven at speeds ranging from 180 r.p.m. to 800 r.p.m., producing thereby a variation in the number of pulsations per second.

15 *Quieting Elements.* Considerable study was made of this essential feature and many trials were made to satisfy ourselves that we were securing pulsationless flow where and when needed. The tank *T*, Fig. 1, was used to quiet the pulsation before the meter station, *M*, was reached. This was a 48-in. vertical tank of 200 cu. ft. capacity. Air could be admitted either at the bottom or at the top and released from the tank through the internal pipe which

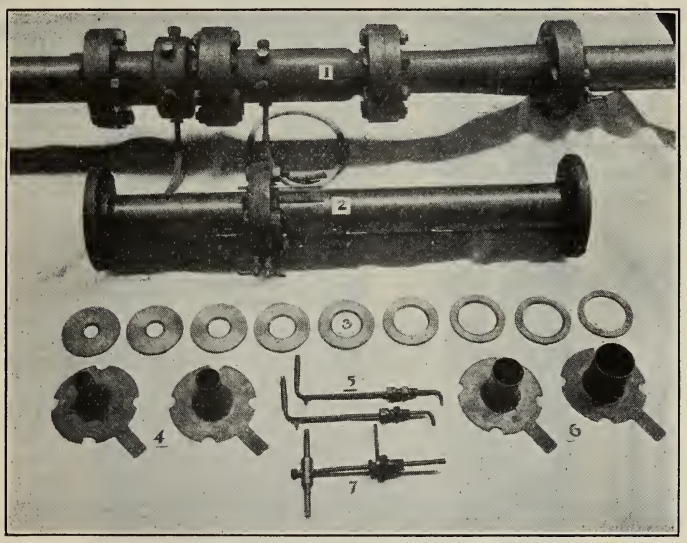

Fig. 3 Flow-Meter Units

reached nearly to the top. This tank when fitted with $1\frac{1}{4}$-in. orifices at top and bottom enabled us to get air flow free from pulsations before the test meters were reached.

16 A second quieting tank was inserted at *Q*, Fig. 1, at a point 8 ft. below the meter station. This was necessary in order to secure pulsationless flow at the orifice head for the purpose of establishing standard flow conditions. This tank was selected almost by chance and afterward was proved by test to be of sufficient capacity to eliminate practically all of the effect of the pulsating flow, and with the insertion of a $1\frac{1}{4}$ in. orifice at the exit from the tank we were entirely successful in securing pulsationless-flow conditions.

17 There are two positions marked, *V*, Fig. 1, one near the compressor and one just beyond the large quieting tank, where volumes of different sizes were inserted for the purpose of studying their quieting effect. Because the term volume seems to apply better we have perverted the word " volume " from a term meaning capacity to a special designation, and have used it altogether to denote the various tanks of different dimensions which have been

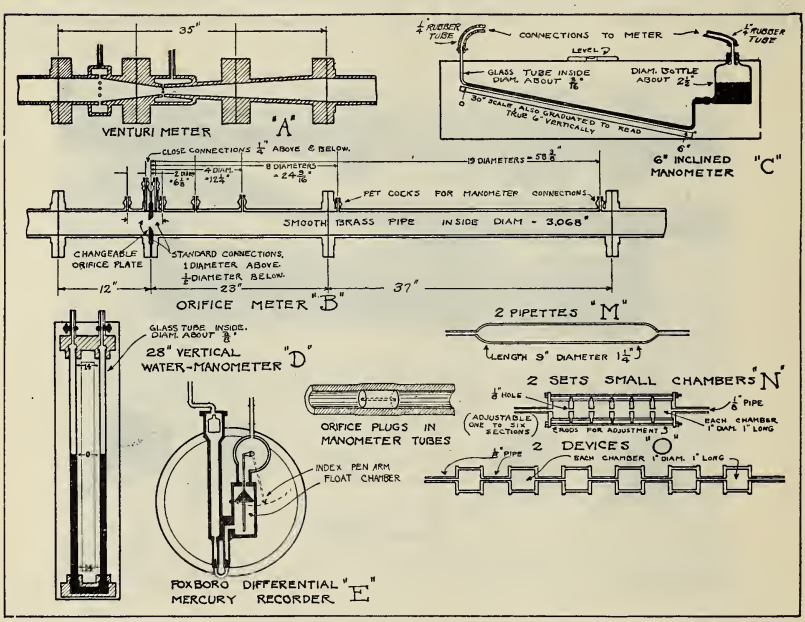

Fig. 4 Section Sketches of Meters, Manometers, and Connections

used as quieting elements. Most of these volumes were used in the second volume space, *V*, beyond the large quieting tank.

18 The line valve near the entrance to the 3-in. test line was used as a quieting element when employed as a throttling device. Both a gate valve and a globe valve and in some cases orifices were thus used as throttling devices.

19 The combination of throttling with volumes constitute the muffler type of quieting device. Fig. 7 shows an 8-section pipe-flange muffler which was also inserted in the line at the second volume station. This muffler is located in a by-pass in front of the line which runs directly from the compressor. All the volumes used

as quieting devices were placed in direct line, but later it was found that the by-pass position answered just as well for the muffler or the other quieting devices. Fig. 9, Nos. 1, 2, 3, 5, 6, shows some of the disks which were tried in the muffler. Fig. 8 shows sectional views of the pipe-flange muffler and also other forms of mufflers, including two funnel mufflers. A form of pulsating bag, Fig. 6, was used as a quieting volume and was connected to the compressor

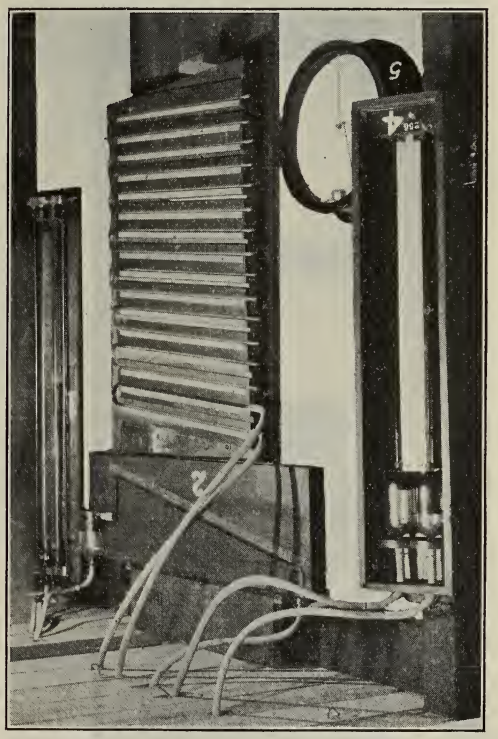

Fig. 5 Manometers Used with Flow-Meter Units

line in a way similar to an air chamber on a reciprocating pump. Another form of device used was a system of revolving fans or baffles. (See Fig. 9, No. 7.)

20 *Measuring Elements.* Under this heading will be taken up in order: (1) the orifice-head meter, (2) meter elements used, and (3) manometers used.

21 *Orifice-Head Meter.* It was recognized at the outset that one of the indispensable features was an accurate method of measur-

ing the discharge of the pipe, or (its equivalent) an accurate means of indicating the velocity of the air in the pipe. This was effected by means of an orifice head at the end of the pipe line, Fig. 1, *H*. Fig 2 is a detailed sketch of the orifice head. Its outer diameter is 9 in. for a distance of 2 ft., followed by a tapered section to meet the 3-in. line. This is given a taper of 7 deg., and was so chosen

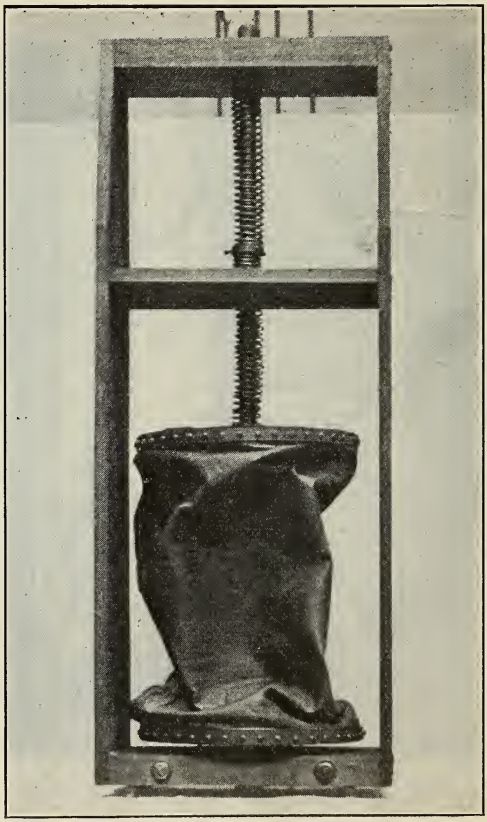

Fig. 6 Pulsating Bag

as being the limiting[1] angle for preventing as far as possible the swirling and eddying of the air as it passes into the orifice head from the line. Seven holes, reamed to $1\frac{1}{8}$ in., were provided in the head plate ($\frac{3}{16}$ in. thick), although during the tests not more than

[1] Trans. Am.Soc.M.E., vol. 42, 1920, p. 26: Physical Basis of Air Propeller Design, F. W. Caldwell and E. N. Fales.

five holes were used at one time. In general the capacity of the compressor was reached with four holes open with a standard static discharge head of 0.9 in. of water at the orifice head. The orifice head was calibrated by discharging air through it from the calibrated tank T under a constant static head at the orifice head. Readings every fifteen seconds were taken of this static head at the orifice head, giving 25 to 45 readings per run. The formula for a single orifice is:

$$q = KAV = 0.1261\ K \sqrt{h/W_a}$$

where q = air flowing cu. ft. per sec.
K = coefficient of discharge
h = static head at orifice head, inches of water
W_a = weight of 1 cu. ft. of air at temperature of flowing air at orifice plate.

For separate, single orifices the maximum value for $K = 0.6269$, the minimum, $K = 0.6037$; average for 13 runs, $K_m = 0.6086$. For orifices in combinations of twos, threes, fours, and fives, average of K for 22 runs including the single orifices = 0.6115.

22 The orifice head was also checked against a second orifice to show how uniformly the air was distributed in its cross-section. A variation was found between the center orifice and the surrounding single orifices ranging from 0.1 per cent below to 0.2 per cent above the ratio for the center orifice. The perforated baffles were then put in. These were two in number, the first being made up of a $\frac{1}{4}$-in. wooden strip screen with $\frac{1}{4}$-in. openings. Two inches in front of this wooden screen was placed a second perforated iron plate with 600 $\frac{1}{4}$-in. holes (see Fig. 2). All possible combinations of the orifices including single orifices and 5-hole orifices with and without the center orifice gave a variation not exceeding 0.1 per cent. This established uniform flow conditions in the orifice head regardless of the number of orifices open in the head plate.

23 For the standard 3-in. pipe (inside diameter, 3.068 in.) with a manometer head at the orifice head of 1 in. of water, barometer 29.50 in., humidity at 75 per cent, the velocity of air in the pipe for each orifice has been calculated as follows:

Temperature of air, deg. fahr.	Velocity of air in pipe, ft. per sec.
50	5.49
60	5.55
70	5.61
80	5.67
90	5.73

Wherever mention is made of the velocity of air in the pipe we have assumed 5.35 ft. per sec. for each orifice for a standard orifice head of 0.9 in. of water at 75 deg. fahr.

24 *Meter Elements Used.* The point of insertion of the meter elements in the line, Fig. 1, *M,* was about 70 ft. from the compressor, 50 ft. above the orifice head, and 25 ft. from the entrance to the 3-in. test line.

25 The venturi meter was a standard unit with 3-in. entry and 1-in. throat. It is shown attached to the pipe line in Fig. 3, No. 1, and in sectional view in Fig. 4, *A*.

26 The orifice meter, Fig. 3, No. 2, was made up of a flanged section of 3-in. brass pipe of the same length, 35 in., as the venturi section. The orifice flange, Fig. 4, *B,* was placed 12 in. from the upstream end and was counterbored to receive the set of orifice plates and to center them accurately. The downstream side of the hole in each plate was chamfered to $\frac{1}{32}$ in. in thickness, Fig. 3, No. 3.

Ratio of orifice and pipe diameters, per cent	Size of orifice plates: Diameter of orifice, in.
33	1.098
40	1.224
50	1.530
60	1.836
70	2.142
80	2.448
90	2.754
100	3.060

The manometer connections were made at one diameter above and one-half diameter below the orifice plate as the standard points of connection.

27 The flange nozzle meter was made by inserting in the orifice-meter section a special-shaped rounded-edge orifice with projecting cylindrical end, as in Fig. 3, Nos. 4 and 6. This nozzle

approximates the converging part of the venturi tube in shape. The manometer connections were made at one diameter above and one-half diameter below the flange, corresponding to the standard points of connection of the orifice meter.

28 Two forms of pitot tips were used. Fig. 3, No. 5, shows the hatchet-edge static tip (pitot No. 1) with $\frac{1}{8}$-in. side openings. This is used with the accompanying open-ended impact tip with $\frac{3}{16}$-in. opening. Both tubes are made of $\frac{1}{4}$-in. seamless brass tubing. Fig. 3, No. 7, shows a modified form of pitot tip (pitot No. 2) having $\frac{5}{16}$-in. brass tubing and an impact or leading opening facing

TABLE 1 SIZE OF FLANGE NOZZLES

Diam. of nozzle, in.	Inlet diameter at rounded edge, in.	Length, in.		
		Rounded edge	Cylindrical end	Total
$1\frac{1}{8}$	3	1	2.375	3.375
$1\frac{1}{4}$	3	1	2.375	3.375
2	3	1	2.375	3.375
$2\frac{1}{2}$	3	$\frac{5}{8}$	2.750	3.375

the direction of flow and a static or trailing opening directly opposite. The diameter of each opening is $\frac{1}{8}$ in.

29 *Manometers Used.* As far as possible the simpler forms of manometers were used. The vertical U-tube water manometer, as in Fig. 5, No. 1; Fig. 4, *D;* Fig. 10, *S,* was used with the venturi meter, and part of the time with the orifice meter and flange nozzle meter, and also for the static line pressure. Where the readings had to be magnified, use was made of a 6-in. inclined one-leg reservoir oil manometer, 5 to 1 magnification (Fig. 5, No. 2; Fig. 4, *C*). For certain other readings an inclined U-tube oil manometer and a vertical U-tube two-liquid manometer were used (Fig. 5, Nos. 3 and 4).

30 A Foxboro differential mercury recording gage, Fig. 5, No. 5 and Fig. 4, *E,* was used to make comparisons with the water manometer used with the venturi meter. The flow conditions were maintained and checked at the orifice head by means of an Ellison inclined gage of 1 in. range and 10 to 1 magnification.

NATURE OF THE PULSATION

31 The first knowledge of the nature of the pulsation was gained through the use of an instrument which we have called the "photopulsometer." This instrument was made and loaned to us by Mr. H. N. Packard of the Cutler-Hammer Co., Milwaukee, Wis. Its principle is similar to that of the "phonedeik" designed by Prof. D. C. Miller[1] of Case School of Applied Science and used by him in photographic studies of musical sounds.

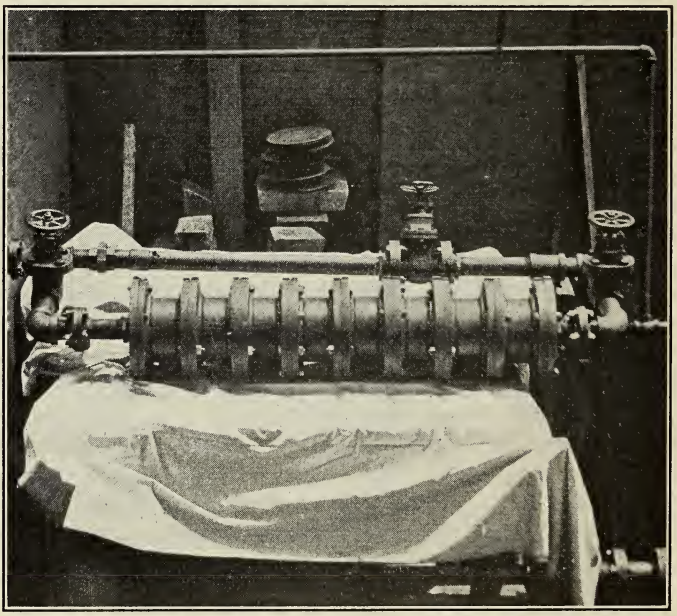

Fig. 7 Eight-Section Pipe-Flange Muffler

32 The photopulsometer is shown in the diagrammatic sketch of Fig. 11. A pitot tube with one leading and one trailing tip set with the opening in line on a vertical diameter was inserted in the center of the pipe. The leading or impact tip communicated with the under side of a diaphragm chamber. The trailing or static tip led to the upper side of the diaphragm chamber. The mica diaphragm, 0.011 in. thick, would therefore respond to changes in velocity of the air in the line as they occurred. These vibrations

[1] The Science of Musical Sounds, p. 78, D. C. Miller.

were directly transmitted to a mirror hung in jeweled bearings and by means of a beam of light could be thrown on a photographic film giving a diagram proportional to the velocity. By means of a pendulum beating quarter- and half-seconds a chronographic record could also be made as shown on most of the films, for example in Fig. 14, by the breaks in the diagrams. A great many films were taken in this manner under a number of different running conditions and it proved to be a valuable method for providing a permanent record of the state of the flowing air in the line, either under violent pulsations due to various disturbing factors or for

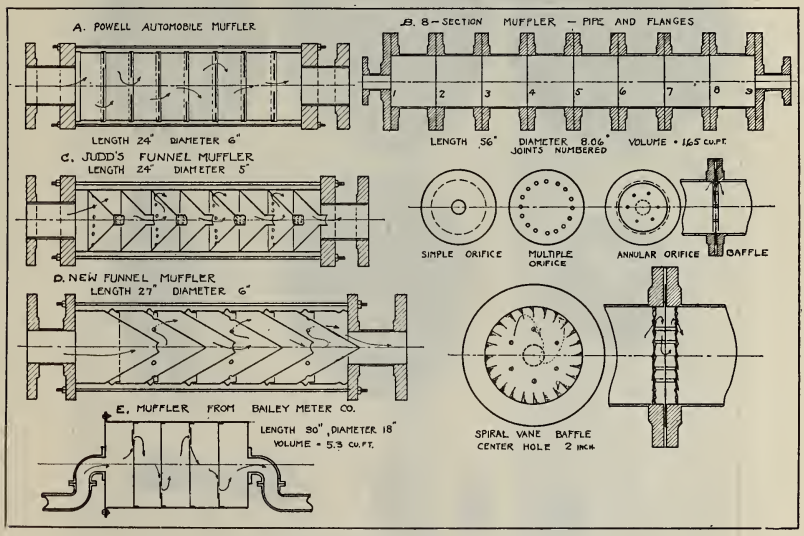

FIG. 8 MUFFLER DEVICES

more steady flow due to the effect of certain quieting factors, as well as a record of the state of flow when under steady or pulsationless-flow conditions.

33 *Velocity Diagrams.* Figs. 12 to 19, inclusive, give records of the velocity changes at the center of the pipe line at various points and under various flow conditions. The maximum effect of the pulsation in the open line direct from the compressor is shown in Fig. 12. There is little or no quieting effect due to a length of pipe equal to 183 diameters. As measured from the diagrams:

Average of maximum pulsation at 50 ft. from compressor = 1.05
Average of maximum pulsation at 111 ft. from compressor = 0.80
Square root of ratio of 80 to 105 = 0.875.

This equals 12.5 per cent quieting effect due to length of pipe for a maximum error of 82 per cent due to pulsating flow.

34 Fig. 13 shows artificial pulsations as produced by the butterfly-valve interrupter and that they are much less violent than those shown in the previous figure where the valve to the compressor is an automatic valve with a light spring. Diagrams (*a*)

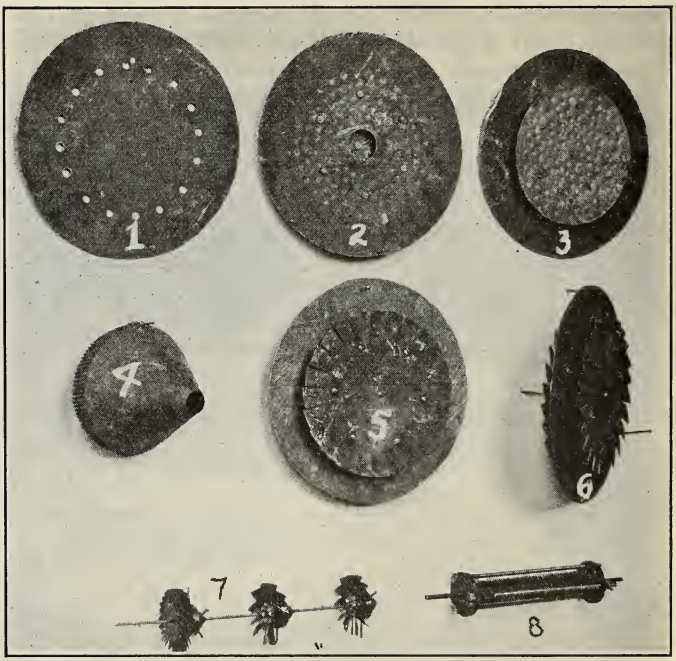

Fig. 9 Muffler Disks, Fan Units, and Small Quieting Devices

and (*b*) approach more nearly the shape of a sine curve. It should be noted that whenever the interrupter was used the air supplied came from the tank *T*, Fig. 1, used as a storage tank. The air was therefore free from pulsations except those that were imparted by the interrupter itself.

35 In Fig. 14 are shown artificial pulsations which are most violent near the interrupter. Those from a point 4 ft. [Diagram (*a*)] above the interrupter show only a slight disturbance indicating that the pulsation is retarded somewhat by the flowing current of air. Diagram (*d*), taken beyond the 24-in. volume,

shows its quieting effect and also shows the kind of diagram to be expected for pulsationless-flow conditions.

36 Diagram (*a*), Fig. 15, also shows pulsationless-flow conditions when the violent pulsations direct from the compressor have been eliminated by the use of the big tank *T* as a quieting device.

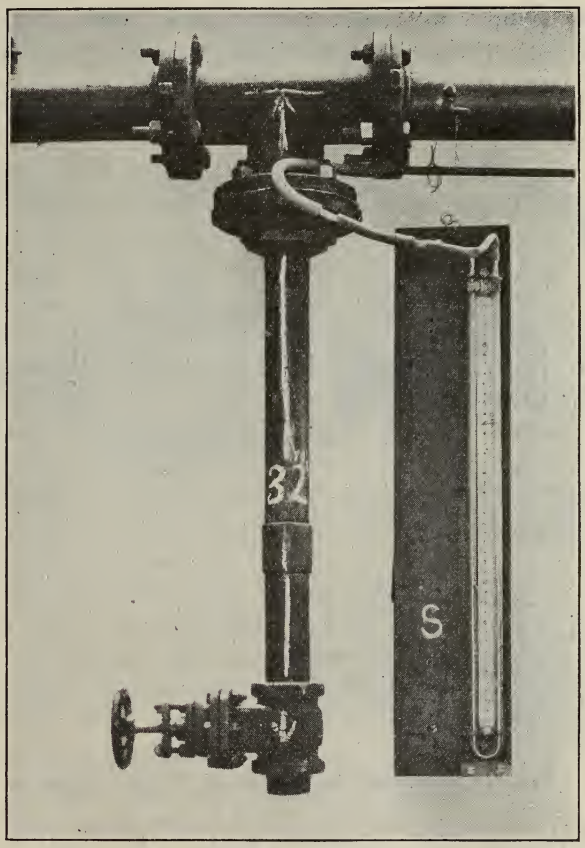

Fig. 10 Manometer for Pressure in Line

Diagram (*b*) shows that the diaphragm of the photopulsometer is not affected by the sound waves from a phonograph. Diagram (*c*) was taken to determine the natural period of vibration of the photopulsometer. This shows 6 vibrations in 0.103 sec., or 58 vibrations per second. This will explain the minute secondary pulsations occurring in Diagram (*a*) and in others under pulsationless

flow. Most of the secondary pulsations seen in the other diagrams are likewise due to the same cause. Due to this cause also, some of the diagrams appear to go below the zero line when in reality they do not. Some other cases arise, however, where the pulsation appears to be negative, for which no good reason could be assigned.

37 The large tank T was tried as a quieting chamber, when connected similar to an air chamber to a pump. Fig. 16 shows that such an arrangement is worthless as a quieting device. This

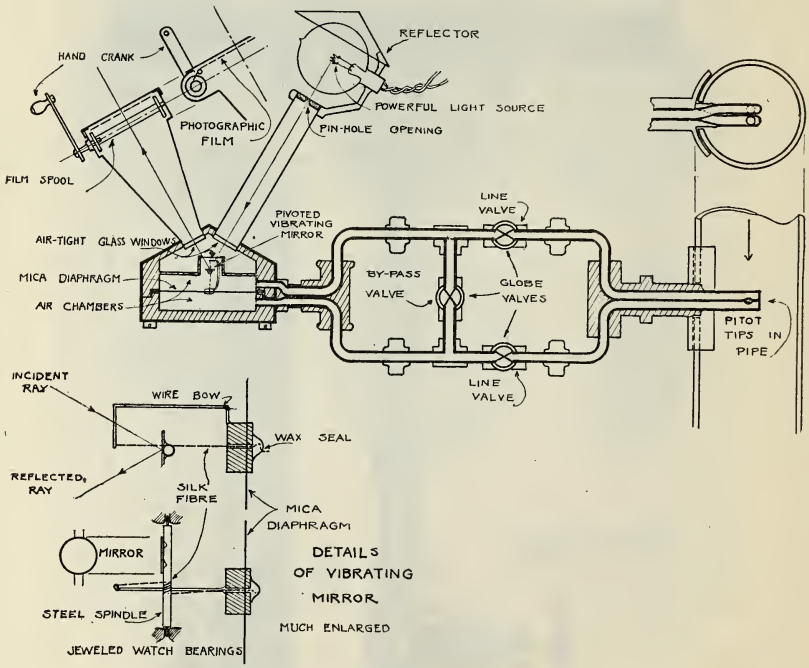

Fig. 11 Diagrammatic Sketch of Photopulsometer

substantiates our later experience that, to be effective, tanks, or volumes, should be inserted in the line so that the air may pass through them axially.

38 One of the methods for eliminating pulsations is throttling by means of some kind of obstruction in the line such as a valve or an orifice. Diagram (a), Fig. 17, is taken for maximum pulsations direct from compressor. The diagrams appear to go below the zero pressure line, but when the secondary pulsations due to the natural period of vibration of the diaphragm are considered, it

will be seen that the true diagrams approach but do not go below the zero line. Diagram (*b*) shows the quieting effect of an orifice when the pulsation has been reduced to the condition of pulsationless flow.

39 *Effect of Pulsation of the Static Pressure in the Line.* For a compressor speed of 293 r.p.m., or nearly five revolutions per second, and for an air velocity in the pipe from 22 to 27 ft. per sec., the velocity wave length as shown by the diagrams would be from 4 to 6 ft. On starting the compressor it seemed frequently that the first impulse traveled much faster than the actual velocity of the air.

40 *Pipe-Line Pressure Diagrams.* To test this out and also to study the effect of pulsation on the static pressure in the line, two Crosby indicators were attached to the line, one on the compressor cylinder (Fig. 1, *C*), and the other 70 ft. distant (Fig. 1, *A*, No. 1 indicator) near the meter space. For details of arrangement of indicators for taking simultaneous pressure readings, see also Fig. 1, *B*. Diagrams from the compressor cylinder (Fig. 20, *A*, No. 1) had been taken several times before this as had also diagrams on indicator at point No. 1, Fig. 1, to show the rapid fluctuation in static pressure. When the indicator at point No. 1 was moved by hand a diagram was obtained closely resembling that shown on the films from the photopulsometer; but no diagrams had been taken simultaneously.

41 Since the sudden rise in the diagram taken by indicator No. 1 must mark the *beginning* of a pulsation, it was concluded that this initial point was the point at which the valve opened on the compressor, thus releasing a pulsation into the air line. This pulsation would continue with the upward stroke of the piston and at the instant that the valve on the compressor closed the pulsation on the record would cease.

42 The two indicator drums were connected by means of a light piano wire to the reducing motion on the engine. The two indicator pencil motions were connected by an electric circuit operating a detent motion. The pencil motion on the indicator at the compressor closed a switch and by means of a solenoid operated the pencil motion of the second indicator. Whenever simultaneous cards were taken, this method was used.

43 *Pressure Pulsation Greater Than Velocity Pulsation.* Three sets of such diagrams were taken and are shown at *A* and *B*, Fig. 20. Two important features were brought out: (1) The pulsa-

tion in the pipe produced a much greater pressure effect than that imparted to the velocity of flow. The diagram taken with the pitot tip (Fig. 20, *B*, No. 2) would be expected to give the combined effect due to pressure and velocity; and a larger and somewhat modified diagram might be looked for. On the contrary, the two diagrams taken with the static connection and with the pitot tip are strikingly similar in shape. (2) When simultaneous diagrams for a single stroke were recorded (Fig. 20, *A* and *B*, No. 3) it was found that the suction stroke of the compressor corresponded to the pressure stroke in the line. This seemed to indicate that the

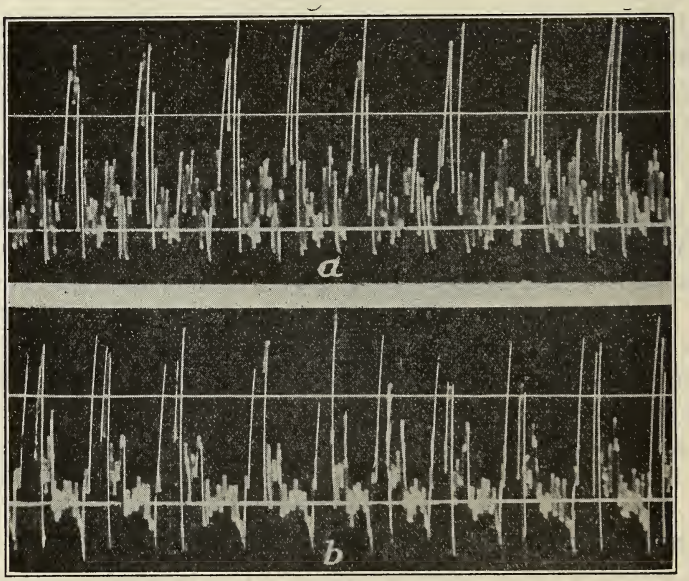

Fig. 12 Showing Maximum Effect of Pulsations at Different Points on the Line — No Quieting Effect Due to Increase in Length of Line

(*a*) Maximum pulsation in line 50 ft. below compressor. (*b*) Do., 111 ft. below compressor.

pulsation required about the time of a compressor revolution to travel a distance of 70 ft. in the pipe. For a speed of 291 r.p.m. this would mean about 0.1 sec. for $\frac{1}{2}$ revolution, or a pulsation velocity of 700 ft. per sec.

44 To verify this assumption, two more indicators were added and their location rearranged as shown in Fig. 1, *A* and *B*. The one at the compressor remained unchanged, a second (at *O*, Fig. 1, *A*) was placed 18 in. distant in the discharge line to establish the

beginning of pulsation (see Cards 14*c*, 14*o*, Fig. 21); indicators I and II (Fig. 1, *B*) were placed in the 3-in. line, 54 ft. and 84 ft., respectively, from the compressor. They were connected to the detent motion through the electric circuit and were arranged so as to be operated either by hand or by the reducing motion. In addition, sparking points were attached to indicators I and II and to the secondary coil, grounded through the pipe line, so that, by means of a commutator, the cards on Nos. I and II would be per-

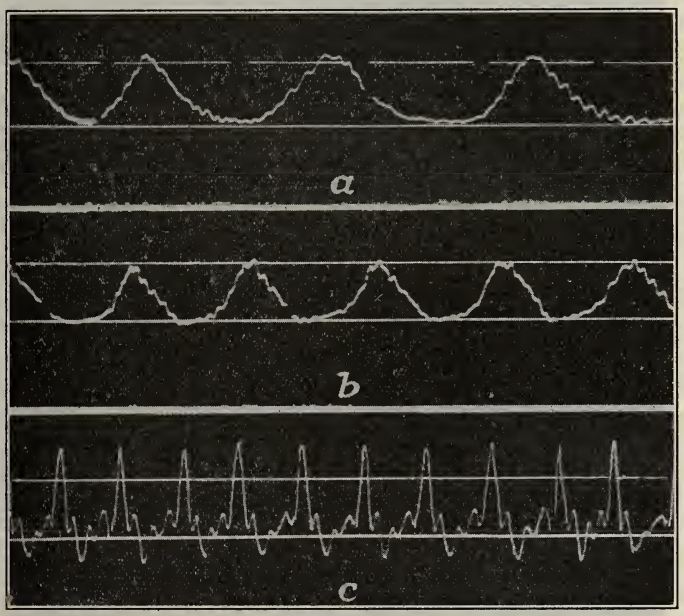

FIG. 13 PULSATIONS PRODUCED ARTIFICIALLY BY BUTTERFLY-VALVE INTERRUPTER. VELOCITY OF AIR IN LINE, 27 FT. PER SEC.

(*a*) 53 ft. below interrupter, 3 pulsations per sec. (*b*) 53 ft. below interrupter, 4 pulsations per sec. (*c*) 53 ft. below interrupter, 13.5 pulsations per sec.

forated simultaneously by twelve sparks for each revolution of the drum while operated by hand. (See Fig. 21, Cards 11_{I}, 11_{II}.)

45 *Analysis of Pressure Diagrams.* Fig. 21 comprises diagrams traced from cards taken on these indicators, including simultaneous cards at points *C* and *O*, *C* and *I*, *C* and II, *C*, I and II, and cards taken at I and II with coincident points shown by spark points. These diagrams include:

a Partial cards for establishing the fact that the initial point

of pulsation is the initial point of pressure rise in the line which is seen to correspond to the point of valve opening in the compressor. (See diagrams to establish simultaneous pressure points, Cards 6*c*, 6*o*, 14*c*, 14*o*)

b Complete diagrams plotted in one direction, showing the use of the coincident lines. The drums were moved by hand and,

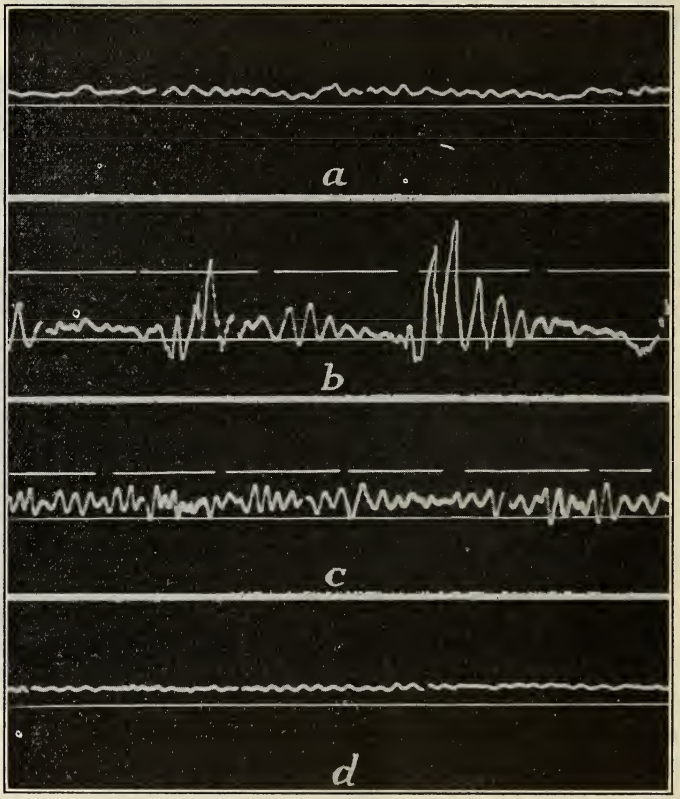

Fig. 14 Showing Artificial Pulsations at Different Points in the Line

Distances from butterfly-valve interrupter: (*a*) 4 ft. above; (*b*) 0.5 ft. above; (*c*) 2 ft. below (*d*) 60 ft. below, showing quieting effect of 24-in. volume.

at several points, simultaneous lines were drawn by means of the detent motion. This gives a method of accurate comparison of selected pulsations.

c Also hand-operated diagrams to illustrate the initial pulsation points for repeated pulsations. The identity of the simulta-

neous strokes on the indicators is established and shown by means of the arrows of direction. (See Cards 3*c*, 3*o*.)

46 For the hand-operated cards the simultaneous lines are numbered alike and the initial points on the same pulsation at different points as it progresses are lettered *AAA*, *BBB*, *CCC*, etc. The progress of the pulsation down the pipe is shown clearly and the similarity of the shape as compared with the record from the photopulsometer is also apparent. (See Cards 3*c*, 3_{I}, 3_{II}; 11_{I}, 11_{II}.)

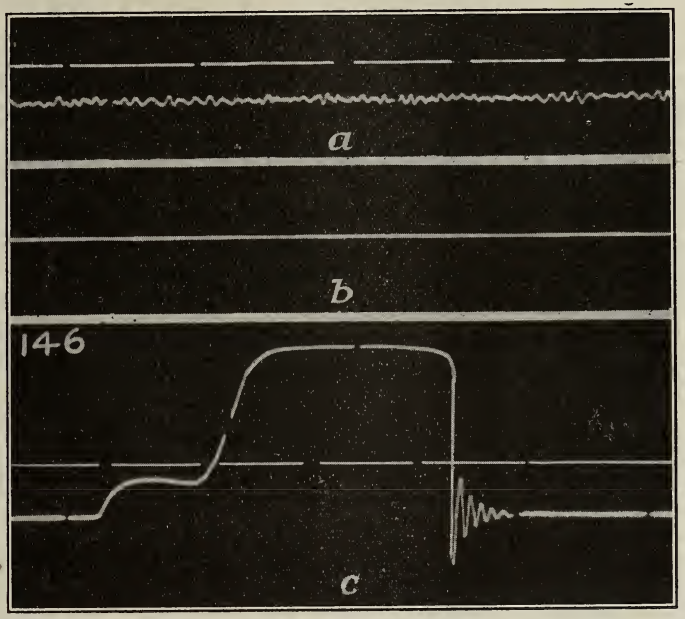

Fig. 15 Showing Velocity Conditions for Pulsationless Flow, and the Natural Period of Vibration of the Diaphragm

(*a*) Taken 111 ft. below compressor; air flowing from tank, no pulsations; velocity of air in line, 27 ft. per sec.

(*b*) Taken with tips placed in front of phonograph, sound concentrated in convergent nozzles.

(*c*) Showing the natural period of vibration of the photopulsometer.

47 *Rectification of Pressure Diagrams.* A few diagrams have been plotted to a uniform horizontal scale, or in other words, the diagrams have been converted into pressure-time diagrams. This method is clearly shown in Fig. 22. The motion of the piston has been considered as that of simple harmonic motion, which is not strictly true, owing to the finite length of the connecting rod. By plotting the complete pulsation as if the diagrams were taken in

TABLE 2 VELOCITY OF PULSATION IN PIPE

COMPUTATIONS OF VELOCITY OF PULSATION BETWEEN INDICATORS I & II

Card No.	Coincident line	Pulsation measured	Length of pulsation in 100ths of an inch	Value of 0.01 in. in seconds of time	Distance from beginning of pulsation to reference line		Lead I–II 100ths of an inch	Time taken by pulsation to travel 30 ft.	Velocity of pulsation, ft. per sec.	Number of holes at orifice head	Remarks
					I	II					
10	4	C–D	72	0.00285	38	28	10	0.0285	1052	4—3.4.6.7.	
10	6	D–E	73	0.00281	43	34	9	0.0253	1185	4–	
10	8	E–F	62	0.00330	48	40	8	0.0264	1135	4–	
10	9	F–G	63	0.00325	26	17	9	0.0292	1028	4–	R.p.m. = 293
11	3	A–B	74	0.00277	24	13	11	0.0305	985	3–4.6.7.	
11	5	B–C	70	0.00293	28	17	11	0.0323	930	3	
11	7	C–D	55	0.00373	34	28	6	0.0224	928	3	
11	10	E–F	53	0.00387	45	38	7	0.0271	1106	3	

COMPUTATIONS OF VELOCITY OF PULSATION BETWEEN INDICATORS C, I & II

Card No.	Coincident line	Pulsation measured	Length of pulsation in 100ths of an inch	Value of 0.01 in. in seconds of time	Distance from beginning puls. to line			Lead in 100ths of an inch			Time taken by pulsation to travel			Velocities, ft. per sec., based on distance		
					C	I	II	C–I	C–II	C–	54 ft.	84 ft.	30 ft.	54 ft.	84 ft.	30 ft.
5	2	A–B	145	0.00141	24	3	–10	21	34	13	0.0296	0.0480	0.0183	1830	1950	1640
5	6	B–C	122	0.00168	52	15	–4	37	56	19	0.0662	0.0941	0.0319	869	893	941
6	3	A–B	120	0.00171	42	21	5	21	37	16	0.0359	0.0632	0.0274	1510	1330	1095
6	6	B–C	110	0.00186	42	18	4	24	38	14	0.0446	0.0706	0.0261	1210	1190	1148
7	3	A–B	150	0.00136	32	–3	–18	35	50	15	0.0478	0.0683	0.0205	1130	1230	1465
7	7	B–C	105	0.00195	42	7	–12	35	54	19	0.0683	0.1052	0.0371	791	797	809

one direction only, the true characteristic shape of the pressure diagram is obtained. When the pulsation is foreshortened and repeated it bears a striking resemblance to those shown on the hand-operated diagrams in Fig. 22 and shown also on the film of the photopulsometer.

48 *The Velocity of the Pulsation.* The determination of the velocity of pulsation from the simultaneous sets of indicator diagrams for points I and II located 30 ft. apart was made as follows:

a Hand-operated diagrams were taken as previously explained.

b The diagrams were carefully compared, points of spark puncture located, coincident lines drawn thereform and numbered, and the initial points of the pulsations were identified and marked *AAA, BBB, CCC,* etc. The unavoidable irregularity of hand operation rendered the identical pulsations easily recognizable by means of their relative lengths.

c Measurements of the sets of diagrams were carried out, as shown in Table 2. It will be noticed that the computations shown include some of the diagrams illustrated in Fig. 21. All the diagrams shown are tracings of originals and are numbered and dated.

d Results were computed as shown on Table 2. Choosing Diagram 11 (see Fig. 21, Cards 11), coincident line No. 3, pulsation *A-B* is 0.74 in. long. For 293 r.p.m. one revolution = 0.205 sec. = time of one pulsation. If 0.74 in. or one pulsation = 0.205 sec., 0.01 in. = 0.00277 sec. Point *A* is 0.24 in. from line 3 for indicator I, and is 0.13 in. from line 3 for indicator II, a lead at point I over point II of 0.11 in., or the time taken to travel 30 ft. will be 0.00277 x 11 which is 0.0305 sec. Rate of travel per second = 30 ÷ 0.0305 = 985 ft., which is the velocity of pulsation. The velocity of sound in dry air at 0 deg. cent., 766 mm. pressure, is 1086 ft. per sec.

49 Table 3 gives the average results computed as in the example given and includes data taken for three points in the line, 30 ft., 54 ft. and 84 ft. from the compressor, and for six different velocities of air flow.

50 These results establish three significant facts:

a That, although our method shows results varying from the maximum to the minimum through a wide range, yet in no case

does the velocity of pulsation so determined approach anywhere near the velocity of the flowing air.

b That, for a variation of velocity of flowing air ranging from zero to 27 ft. per sec., the velocity of pulsation was found to be independent of the velocity of the air.

c That the total average for 148 computations gave 1090 ft. per sec. as the velocity of pulsation. The velocity of sound in dry air at 32 deg. fahr. and 29.92 in. barometer is 1083 ft. per sec. The velocity of pulsation in all probability is equal to that of sound in air.

TABLE 3 AVERAGE RESULTS FOR VELOCITY OF PULSATION

Date	No.	Number of diagrams measured	Length of pipe used, ft.	Velocity of air per sec., ft.	Velocity of pulsation, ft. per sec.		
					Max.	Min.	Avg.
4-5-22	1	14	30	0	1462	640	1042
	2	16	30	5	1412	869	1088
	3	18	30	11	1440	805	1124
	4	20	30	16	1263	775	1019
	5	20	30	22	1290	815	1044
	6	13	30	27	1465	747	1100
	Total	101				Average...	1070
4-7-22	7	16	30	22	1640	809	1143
	8	15	54	22	1830	780	1107
	9	16	84	22	1750	797	1149
	Total	47				Average...	1133

51 For the average velocity of pulsation equal to 1090 ft. and for 4.88 pulsations per second, the pressure wave length as shown on the diagrams would be 223 ft.

52 Since the velocity of the pulsation is independent of the velocity of the flowing air and is evidently equal to the velocity of sound in air, it seems quite reasonable to conclude that the pulsation is a pressure change in the form of a wave front resembling a sound wave of low frequency. It seems also highly probable that these pulsations are similar in character to the pulsations set up by water hammer in a pipe line, since they also travel with the velocity of sound in water.

53 We gained some additional knowledge of the nature of the pulsation by noting its effect on manometers. Where a manometer was used to measure a differential head at a meter, the following effects were consistently present:

a For pulsationless flow the reading was very constant, the only variation being a slight long period surge due to appreciable variations in the compressor speed.

TABLE 4 OBSERVED DATA AT METER

No.	Setting of Line	Instrument Used	Readings on Manometer 1 Lt.	1 Rt.	1 Tot.	2 Lt.	2 Rt.	2 Tot.	3 Lt.	3 Rt.	3 Tot.	Remarks
1	9″×9″ vol. at com-	Vent. 28″ Man.	12.9	13.2	26.1	12.8	13.3	26.1	12.7	13.3	26.0	Temp 2
2	pressor	" Foxboro	cha	rt	19.			19.			19.	= 86.5°
3		indicator card										R.p.m.
4		Film No. 255	at	Ia_2								=288
5	9″×9″ vol. at comp.	Vent. Foxboro			21.8			21.8			21.8	
7	9″×18″ vol. midway	" 28 M.	13.5	13.8	27.3	13.3	13.8	27.1	13.2	13.5	26.7	
10	9″×9″ vol. at comp.	" "	11.7	12.1	23.8	11.7	12.2	23.9	11.7	12.2	23.9	
11	9″×27″ vol. midway	1 " Foxboro			20.6			20.6			20.6	Temp.
14	9″×9″ vol. at comp.	" "			20.0			20.0			20.0	= 99°
15	9″×36″ vol. midway	" 28″ M.	12.8	13.2	26.0	12.7	13.1	25.8	12.7	13.2	25.9	
18	9″×9″ vol. at comp.	" "	12.9	12.2	25.1	12.6	13.3	25.9	12.1	12.8	24.9	(Runs
19	9″×45″ vol. midway	" Foxboro			19.8			19.8			19.8	made
22	9″×9″ vol. at comp.	" 28″ M.	11.9	12.6	24.5	11.9	12.4	24.3	12.0	12.6	24.6	7–20–21)
23	9″×54″ vol. midway	" Foxboro			18.6			18.6			18.6	
26	9″×45″vol. at comp.	" Foxboro			21.0			21.0			21.0	
27	9″×54″ vol. midway	" 28″ Man.	12.0	12.6	24.6	12.0	12.7	24.7	12.0	12.7	24.7	
2	9″×45″ vol. at comp	Venturi meter with 28″ vertical water manometer	9.8	10.6	20.4	9.8	10.5	20.3	9.8	10.4	20.2	
5	9″×18″ vol. " "		14.1	14.4	28.5	14.1	14.3	28.4	14.2	14.3	28.5	R.p.m.
8	9″×27″ vol. " "		12.4	12.9	25.3	12.5	12.9	25.4	12.6	13.0	25.6	= 288
11	9″×36″ vol. " "		11.3	11.9	23.2	11.2	11.7	22.9	11.2	11.8	23.0	(Runs
14	9″×54″ vol. " "		8.8	9.7	18.5	9.1	9.6	18.7	9.0	9.7	18.7	made
17	Line Clear		15.8	17.0	32.8	15.9	17.0	32.9	15.7	17.0	32.7	7–21–21)
22	Pulsationless		4.1	5.1	9.2	4.1	5.1	9.2	4.1	5.1	9.2	
20	Pulsating — O.L.	1″ or 33% orifice meter with 28″ vertical water manometer	14.0	13.2	27.2	14.0	13.2	27.2	14.0	12.9	26.9	R.p.m.
22	Pulsationless		11.7	11.4	23.1	11.9	11.6	23.5	12.0	11.7	23.7	= 291
29	1–1″ Orifice		12.6	12.1	24.7	12.7	12.2	24.9	12.7	12.1	24.8	
30	2–1″ Orifices		11.8	11.4	23.2	12.5	11.7	24.2	11.5	12.5	24.0	T_2=95°
31	3–1″ "		12.2	11.5	23.7	12.2	11.5	23.7	12.2	11.7	23.9	(Runs
32	4–1″ "		12.3	11.6	23.9	12.1	11.5	23.6	11.6	11.6	23.2	made
33	5–1″ "		12.0	11.5	23.5	12.0	11.0	23.0	12.0	11.5	23.5	8–5–21)
2	1″ Orifice	33% orifice meter 28″ vertical manometer	13.9	13.3	27.2	14.0	13.4	27.4	13.8	13.3	27.1	R.p.m.
3	$\frac{28}{32}$″ "		12.3	12.0	24.3	12.5	12.2	24.7	12.4	12.1	24.5	= 291
4	$\frac{25}{32}$″ "		12.2	12.0	24.2	12.3	12.1	24.9	12.2	12.5	24.7	(Runs
5	$\frac{22}{32}$″ "		12.3	12.0	24.3	12.3	12.0	24.3	12.3	12.0	24.3	made
6	$\frac{19}{32}$″ "		12.4	12.1	25.5	12.7	12.3	25.0	12.7	12.3	25.0	8–6–22)

b For pulsating flow this surge was magnified greatly.

c For pulsating flow there is also a rapid vibration of the water column corresponding to the pulsations and depending in amplitude upon the local conditions at the manometer.

d The most significant characteristic of the readings for pulsating flow was the large increase over that for pulsationless flow. This increase was present for every type of manometer, meter and gage tried. It varied from a few per cent to several hundred per cent under extreme conditions.

THE ELIMINATION OF THE PULSATION

54 The problem of the elimination of the pulsation, or of the effects due to the pulsation, suggested two methods of attack: (1)

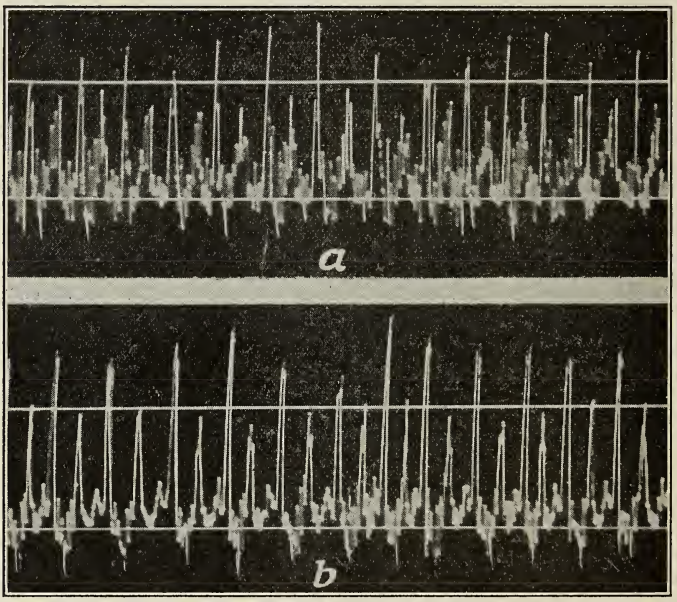

Fig. 16 Vertical Tank of 200 cu. ft. Capacity as Quieting Chamber—Similar to Air Chamber on Pump; 4.9 Pulsations per Sec.

Distances below compressor: (*a*) 50 ft.; (*b*) 117 ft.

modification of the existing metering devices so that the recorded flow would be unaffected whether the flow be steady or in pulsations; and (2) the use of devices which would correct or eliminate the pulsations before the flowing fluid reached the meter.

55 The first of these suggested schemes was taken up to some extent in the study of the modification of manometer connection. The second suggestion, that of pulsation elimination, received the

major part of our attention. Of the five quieting devices used, the pulsating bag and the revolving fan operated by the air flow were studied by means of the photopulsometer. The use of throttling devices, the insertion of tanks, volumes or equalizing chambers in the line and a combination of the two devices, forming the so-called "muffler," comprise the remaining three quieting and eliminating devices. These five schemes, it is believed, cover nearly all, if not all, of the practical schemes which might be used for this purpose.

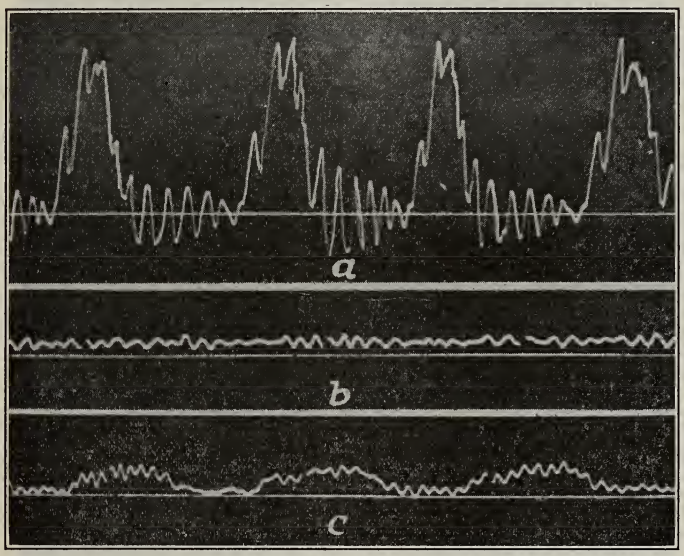

Fig. 17 Showing Quieting Effect of Throttling by Means of an Orifice

(*a*) 80 ft. below compressor, maximum pulsation in line. (*b*) 10 ft. below throttling orifice, pulsation destroyed. (*c*) 4 ft. above interrupter, 3.5 pulsations per sec.

56 The tabulated results which follow include the average observed data and the percentage of error as figured from these average data. The average observed data are the averages of three simultaneous readings taken for all manometer readings. Tables 4 and 5 are observed-data sheets. In most instances where any change in line setting was made a pulsation-flow run was followed immediately by a pulsationless-flow run so as to eliminate the error due to possible change in temperature. In the few cases during the first of the experimentation where this was not done, proper corrections were made later for the temperature effect.

The term " percentage of error " for any meter means the percentage by which the indicated velocity of flow with pulsations differs from the velocity of flow for quiet or pulsationless flow, both read-

TABLE 5 OBSERVED DATA AT ORIFICE HEAD

Run No.	Orifice-head reading			Hydrometer		Instrument or line setting	Remarks
	1	2	3	Dry temp.	Wet temp.		
1	0.911	0.914	0.911	79.0	70.0	Pulsating flow. No. 1. Vol. 9″ × 9″ next to compressor, others from No. 2 to No. 6 at mid point. Venturi meter — alternating Foxboro gage and water manometer.	
2	0.929	0.913	0.916	78.6	69.8		
3	0.909	0.917	0.916	78.9	70.0		
4	0.915	0.909	0.900	79.3	70.1		
5	0.911	0.903	0.905	80.5	70.6		Bar. = 29.047
7	0.917	0.890	0.877	80.4	70.6		
10	0.920	0.916	0.914	81.1	70.0		Zero of orifice-head gage = 0.003 in. (Runs made 7–20–21)
11	0.897	0.892	0.890	81.1	70.1		
14	0.915	0.910	0.891	81.2	70.0		
15	0.878	0.875	0.881	81.0	69.8		
18	0.905	0.903	0.887	82.1	67.7		
19	0.902	0.898	0.901	82.6	67.8		
22	0.895	0.887	0.896	82.8	67.7		
23	0.900	0.895	0.884	83.5	68.0		
26	0.915	0.907	0.894	83.3	67.6		
27	0.905	0.913	0.910	84.0	67.9		
2	0.905	0.906	0.908	74.3	66.5	Open line pulsationless	
5	0.895	0.904	0.898	76.3	66.5		Bar = 29.195
8	0.900	0.893	0.890	77.0	66.6		
11	0.881	0.890	0.903	77.1	66.6		Zero = 0.004 in. (Runs made 7–21–21)
14	0.880	0.882	0.896	77.9	66.8		
17	0.911	0.910	0.910	78.6	66.7		
22	0.894	0.906	0.911	79.3	68.1		
20	0.898	0.902	0.901	79.3	70.1	Open line pulsationless	Bar. = 29.051
22	0.910	0.906	0.913	79.6	71.2		
29	0.913	0.914	0.913	80.0	70.0		
30	0.895	0.920	0.911	80.8	70.5		Zero = 0.006 in.
31	0.915	0.910	0.910	80.5	69.1		
32	0.913	0.917	0.906	80.9	69.8		(Runs made 8–5–21)
33	0.886	0.890	0.882	81.2	71.0		
2	0.910	0.902	0.908	72.5	63.5	1 in.	
3	0.916	0.920	0.919	73.5	69.5	$\frac{28}{32}$ in.	Bar. = 28.98
4	0.915	0.920	0.930	74.0	70.0	$\frac{25}{32}$ in.	
5	0.925	0.928	0.930	—	—	$\frac{22}{32}$ in.	(Runs made 8–6–21)
6	0.930	0.945	0.940	75.0	70.5	$\frac{19}{32}$ in.	

ings taken on the same manometer, or indicating device and with a constant quantity of air flowing under the same conditions. The condition for constant flow was assured by maintaining at all times

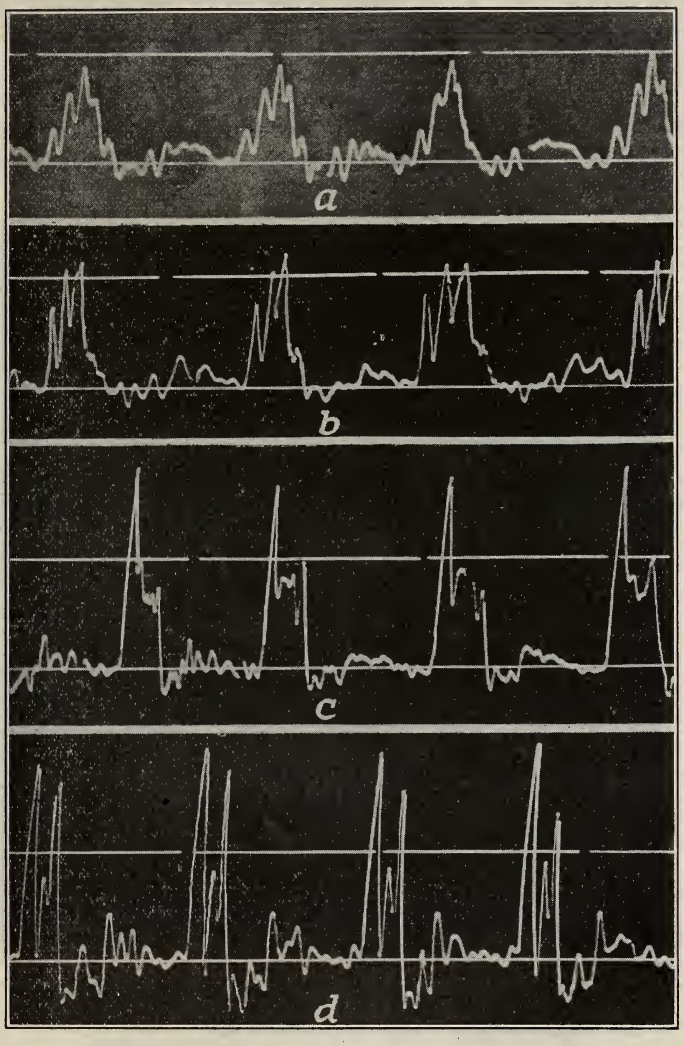

FIG. 18 SHOWING QUIETING EFFECT OF 3-IN. REVOLVING FANS DRIVEN BY CURRENT OF AIR AT VELOCITY OF 22 FT. PER SEC.

(*a*) 54 ft. below compressor, pulsation slightly throttled; maximum error due to pulsation, 68 per cent; 4.5 pulsations per sec.

(*b*) Same as (*a*) but at 80 ft. below compressor.

(*c*) 54 ft. below compressor, 6 ft. above fan section; maximum error due to pulsation, 68 per cent; 4.5 pulsations per sec.

(*d*) 20 ft. below fan section acting as quieting unit; error reduced from 68 per cent to 27 per cent; 4.5 pulsations per sec.

a static head of 0.9 in. of water at the orifice head, where special care was taken to secure pulsationless-flow conditions.

$$\text{Error in Per Cent} = (\sqrt{P_2/P_1} - 1)100$$

where P_2 = corrected manometer reading for pulsating flow
P_1 = corrected manometer reading for pulsationless flow.

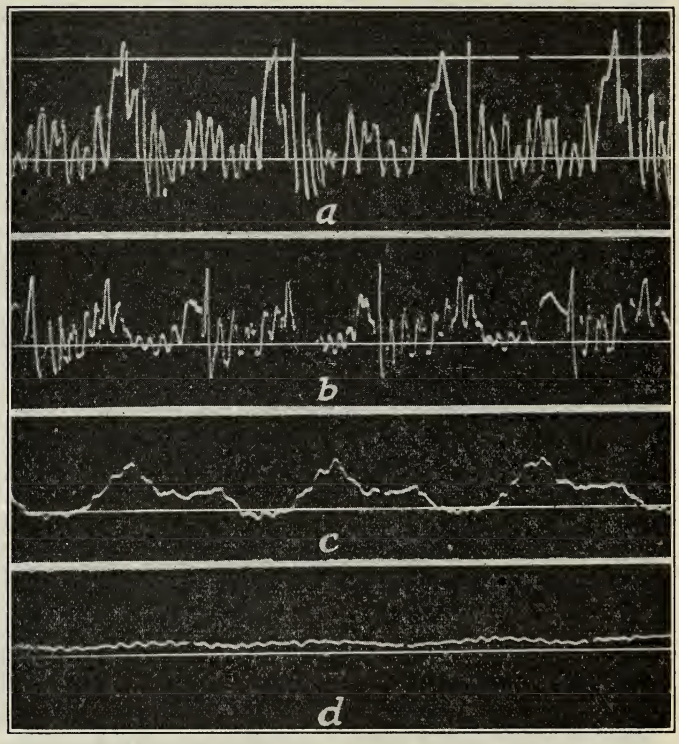

FIG. 19 SHOWING QUIETING EFFECT OF PULSATING BAG

(*a*) 54 ft. below compressor; maximum pulsation in line.
(*b*) 54 ft. below compressor; pulsating bag attached to quiet pulsations.
(*c*) 88 ft. below compressor; shows slight quieting effect of the 12-in. volume.
(*d*) 117 ft. below compressor; shows complete quieting effect of the 24-in. volume.

The error in velocity varies as the square root of the error in head reading. The manometer readings at the meter were corrected to a standard orifice head reading of 0.9 in., by direct proportion.

57 *Modification of Manometer Connections at the Meter.* The first attempt to reduce the error of pulsation by this means

was by throttling the manometer connections. These connections between the meter in the line and the manometer indicating the head readings were made of heavy rubber tubing having an inside diameter of $\frac{3}{16}$ in. It has long been the practice to quiet the swinging of a pressure-gage hand by throttling the gage connections. Following out this idea, a series of " orifice plugs," see Fig. 4, were made by drilling holes of known size in small pieces of brass rod.

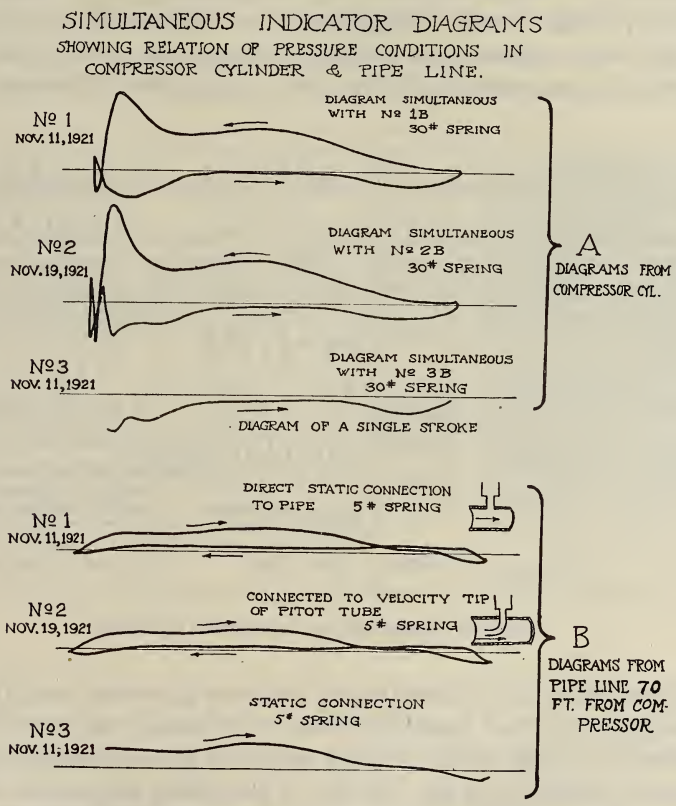

Fig. 20 Simultaneous Indicator Diagrams

These plugs were inserted in the rubber manometer tubes both for the vertical U-tube water manometer and also for the Foxboro differential mercury gage.

58 Fig. 23 gives the average error for both these gages for maximum pulsation while using the venturi meter. The curve shows that throttling has no appreciable effect in reducing the error

until the opening has been reduced to less than 0.07 in. diameter, and that even for an obstruction so small as nearly to close the opening the percentage of error is not reduced to within practical limits. The Foxboro gage showed the same characteristic in regard to the effect of throttling, but gave an error about 60 per cent as large as that for the water manometer. In either case in order to reduce the error to 50 per cent of the maximum it would be necessary to use an orifice of 0.02 in. diameter, or the diameter of a No. 76 drill, clearly a ridiculous size. The surge, or pulsation, of the water column was completely destroyed, so that the effect is quite analogous to that of the steam gage when throttled. This

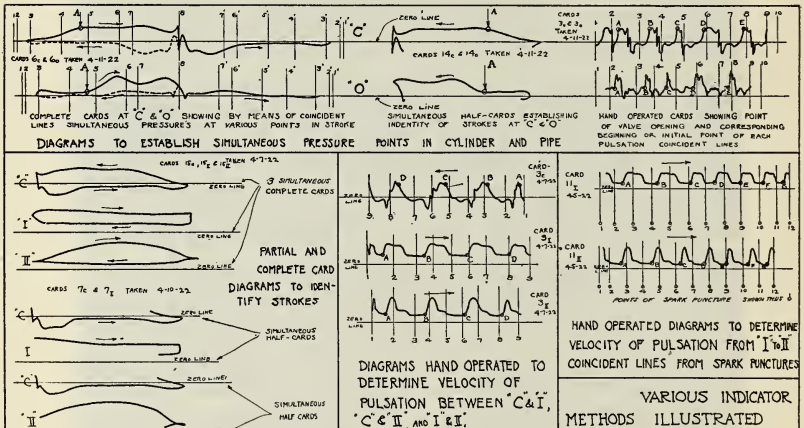

Fig. 21 Analysis of Indicator Diagrams

indicates that while throttling a pressure gage does not affect its reading, it has no beneficial effect in reducing the error due to pulsation. That such an error might exist was pointed out by Mayo [1] in 1905, and the futility of throttling manometer connections to reduce the pulsation is also mentioned by Westcott [2] in 1922.

59 The efficient quieting effect of volumes when used in the test line suggested the possibility that small volumes inserted in the tubes leading to the manometers might serve to reduce the pulsation before the meter was reached.

[1] Trans. Am.Soc.C.E., vol. 54, 1905, Part D, p. 502, Mayo.

[2] Measurement of Gas and Liquids by Orifice Meters, 1922, p. 143, H. P. Westcott.

60 Our first observations as to the possible effect of a volume in the manometer line were made by a comparison of the readings of the two manometers (the 6-in. inclined one-leg reservoir oil manometer and the 28-in. vertical U-tube water manometer) when used interchangeably. These two manometers when used to measure the pulsationless flow would agree exactly. When used for pulsating flow the two would not always agree, although the amount of difference between them was difficult to determine.

61 Several tests were made using the orifice meter and the 6-in. inclined gage with volumes of different sizes in one or both of the manometer connections. The results of these tests show that

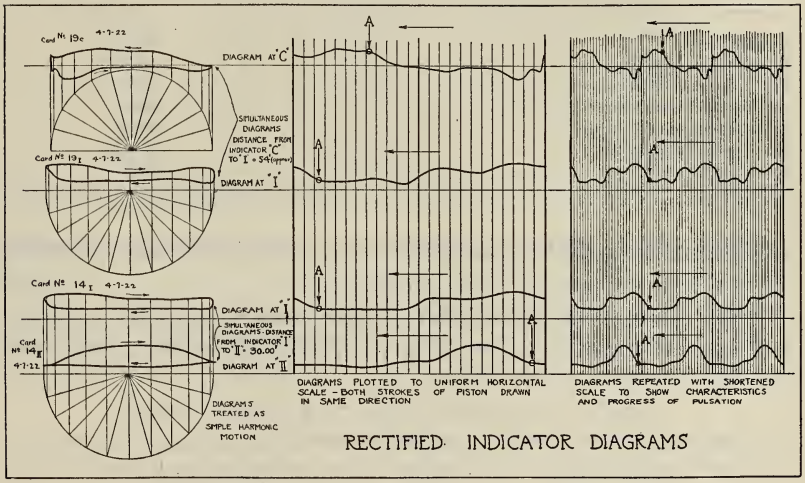

Fig. 22 Analysis of Indicator Diagrams

the use of volumes in the manometer connections does not give so favorable results as the method of throttling. There is only about 15 per cent reduction in the error for the 70 per cent orifice meter.

62 It was thought, also, that the point of attachment of the manometer connection might have some influence on the error due to pulsation. With the orifice meter comparisons were made with the manometer connected (1) close to the orifice and (2) at a distance of one pipe diameter above the orifice and at points below the orifice ranging from $\frac{1}{2}$ diameter to 19 diameters.

63 Tests made show that for all orifices including the 80 per cent orifice the error at the standard points of connection is greater than that for points near the orifice which averages 96.3

per cent of the error at the standard points of connection. For points farther distant from the orifice there is a tendency for the error first to increase and then to decrease as the 19 diameters point is approaching, in all covering a total range of 20 per cent error.

64 *Effect of Type of Manometer Used.* The error in measuring pulsating flow seemed to depend to some degree upon the type

TABLE 6 MAXIMUM ERROR FOR VENTURI, FLANGE NOZZLE, AND PITOT METERS

Kind of meter	Error, per cent, average
Venturi meter	82.0
1⅜ in. flange nozzle meter	76.5
1½-in. flange nozzle meter	94.3
2-in. flange nozzle meter	142.0
2½-in. flange nozzle meter	199.0
No. 1 pitot-tube meter	137.0
No. 2 pitot-tube meter	103.0

of manometer used to register the head, even if all other conditions of the line and meter were identical.

65 It may be stated that all manometers properly graduated will give the same head reading for pulsationless flow. But when

TABLE 7 MAXIMUM ERROR FOR ORIFICE METER

Size of orifice, per cent	Error, per cent, average (from curve)
33	15.0
40	25.0
50	47.5
60	81.0
70	127.0
80	185.0
90	285.0

the flow is pulsating, the ratio of its reading to the true reading will differ somewhat according to the variations mentioned above. A mercury manometer will probably show an error less than that of a water manometer and the latter less than one using mineral oil. A manometer with small tubes is likely to read higher than one with a larger set of tubes, but this is merely a tendency. If the

tubes are too small the capillary effect can be noted; and if they are too large, or if they end in a reservoir, the doubtful effect due to a " volume " will be introduced. The inclined leg of a manometer under some conditions may even cause less " surge " effect. The presence of a check value or damping device between the two manometer legs or chambers, or a float to actuate the recording

TABLE 8 COMPARISON OF METERS

Size, per cent	Error, per cent, average	
	Orifice meter	Flange nozzle meter
37.5	20.0	76.5
50.0	47.5	94.0
66.7	112.5	142.0
83.3	207.5	199.0
	Orifice meter	Venturi meter
33.0	15.0	82.0

arm may reduce the error, as is seen in the case of the Foxboro gage.

66 *Maximum Percentage of Error Produced by Pulsation.* For our installation the maximum error for the meters was as given in Tables 6 to 9, inclusive. The results show that the less the

TABLE 9 RESTORATION OF PRESSURE AFTER PASSING METER[1]

Type of meter	Maximum error, per cent	Restoration of pressure, per cent
Venturi meter	82.0	80.0
33% orifice meter	15.0	11.5
40% orifice meter	25.0	19.0
50% orifice meter	47.0	28.0
60% orifice meter	81.0	38.0
70% orifice meter	127.0	48.0
80% orifice meter	185.0	61.0
90% orifice meter	285.0	77.0

obstruction to the flow of the air, the greater the percentage of error due to pulsation; also, the greater the restoration of pressure after passing the meter, the greater will be the error. The reason

[1] Trans. Am.Soc.M.E., vol. 38, 1916, p. 362, 264: Water Flow through Pipe Orifices, H. Judd.

for this relation appears to be that, since the pulsation is a form of pressure energy, that type of meter unit which in itself most completely dissipates the pulsation energy will show the least percentage of error.

67 *Distribution of Pulsation as Shown by Traverse.* The pipe was traversed by both types of pitot tubes for maximum

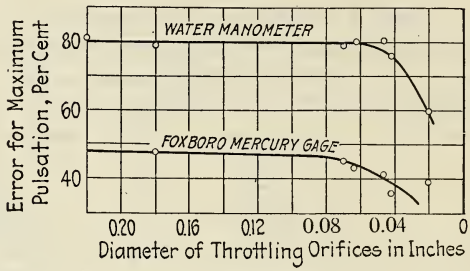

Fig. 23 Effect of Throttling Manometer Connections — for Venturi Meter

pulsation conditions. The results are shown in Fig. 24. The curves of this figure show by both traverses that error due to the pulsating flow is least at the center of the pipe. There is a slight tendency, as shown by the curves, for the point of minimum error

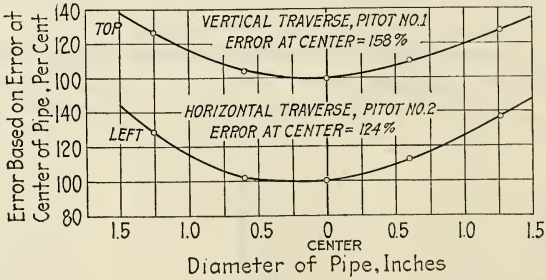

Fig. 24 Distribution of Pulsation Across the Diameter of the Pipe

to be located a little to one side of the center of the pipe. It is not known just how much the pitot tubes themselves are influenced by their approach to the wall of the pipe.

68 *Quieting Effect of a Revolving Fan Section* (see Fig. 9, No. 7). The effect of a revolving fan section when placed in the pipe line is shown by Fig. 18. The revolving fan apparently has

some merit as a quieting device, but is of questionable practical value.

69 *Effect of the Pulsating Bag as a Quieting Device,* Fig. 6, is shown by Fig. 19. It is felt that the special design of such a device would be needed to cover the requirement of each individual installation in order to correct or eliminate the pulsating error.

70 *Elimination of Pulsation by Throttling.* The experiments carried on with the various devices for eliminating or modifying

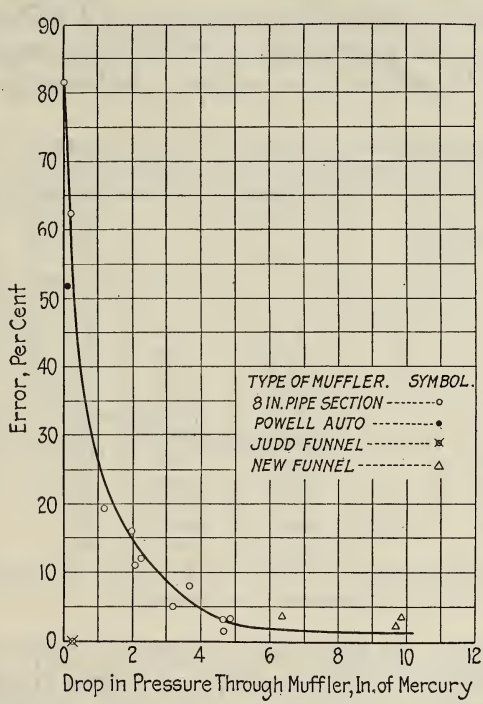

Fig. 26 Effect of Mufflers for Quieting Pulsation — Used with Venturi Meter

the pulsation led to the conclusion that the solution of the problem depended entirely on the absorption of the energy of the pulsation propagated as a pressure wave closely resembling a sound wave of low frequency. Whatever the device used, its value in killing the pulsation will be measured by its ability to absorb, or dissipate, this energy of pulsation.

71 The first attempt made to quiet the pulsation was by means of throttling either by valve or by orifice. The loss of pres-

sure during passage through the valve or obstruction in the line was the eliminating factor.

72 The general effect of throttling is to reduce the error rapidly by means of a pressure drop up to 4 in. of mercury. The use of a greater pressure drop causes the error to be reduced more gradually. The manner of throttling is immaterial whether by gate or globe valve or by an orifice.

73 The general characteristics are summarized in Table 10. This table shows that a drop in pressure of 4 in. by throttling is needed to make an appreciable reduction in the error; and a drop of 6 in. or 3 lb. per sq. in. is necessary to bring the error within

TABLE 10 ELIMINATION OF THE PULSATION ERROR BY THROTTLING

Kind of meter used	Drop in pressure by throttling, inches of mercury				
	0	2	4	9	12
	Error, per cent				
Venturi	82.0	26.0	10.0	5.0	2.5
33% orifice	10.0	2.0	0.0	0.0	0.0
50% orifice	58.0	17.0	7.0	4.0	2.0
70% orifice	145.0	50.0	18.0	9.0	4.0
80% orifice	175.0	67.0	25.0	12.0	4.0
90% orifice	315.0	118.0	22.0	−5.0	18.0
1⅛-in. flange nozzle	44.0	18.0	8.0	3.5	1.0
1½-in. flange nozzle	83.0	28.0	10.0	5.0	1.0
2-in. flange nozzle	156.0	50.0	16.0	6.0	1.0
2½-in. flange nozzle	340.0	97.0	31.0	7.5	1.0
Pitot No. 1	160.0	50.0	20.0	−7.0	2.5
Pitot No. 2	140.0	81 0	5.0	0.0	3.5

practical limits. In most cases the error is not reducible below 1 to 3 per cent, even with a sacrifice of a drop of 12 in. mercury.

74 *Elimination of Pulsation by the Use of Volumes.* Tanks, or volume capacities, or " volumes," as we have chosen to call them, were used in the line for the purpose of quieting the pulsation. These volumes were inserted in the line so that the direction of flow through them was along the axis of the volume. The connections were made by means of pipe flanges; and later orifices were inserted in these flanges at the entrance and exit of each volume. This produced an abrupt entrance into and an abrupt exit out of the volumes, which in itself would tend to cause a loss of energy; and hence would probably contribute toward the reduction of the energy imparted to the pulsation. These volumes were all cylin-

drical in shape and with the exception of the 8-in. and the 48-in. volumes were made of thin sheet metal, No. 24 gage.

75 The quieting effect due to the use of volumes is shown by the curve sheet in Fig. 25. The dotted curve is drawn in as a representative average curve for both venturi and orifice meters where volumes, alone, are used.

76 The 24-in. volume in our test line, situated below the meter, had a capacity of 29 cu. ft., and was plainly one of ample size to convert the pulsating flow into pulsationless flow when the air reached the line leading to the orifice head. It is evident from these results that a volume is also a practical means of eliminating

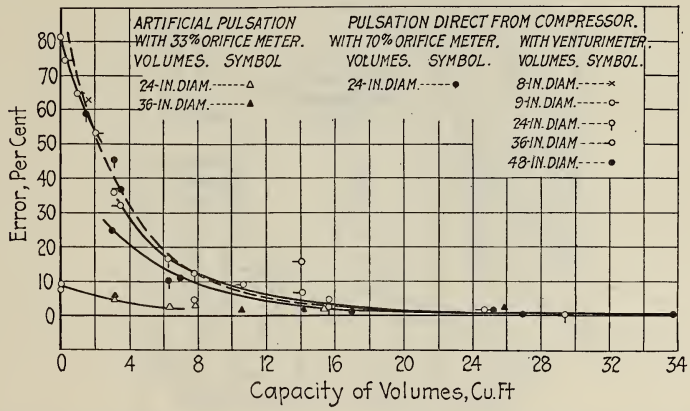

FIG. 25 PERCENTAGE OF ERROR FOR VENTURI AND ORIFICE METERS — PULSATION QUIETED BY VOLUMES

the pulsation. The chief question is, whether in large installations volumes of sufficient size would be of practical use.

77 *Effect of Varying the Shape of Volume* was also studied. In a general way a volume is probably more efficient when it is of relatively large diameter.

78 *Elimination of Pulsation by Combining Throttling with Volumes.* Since the pulsation could be nearly if not quite eliminated either by the use of throttling devices or by the use of volumes alone, the natural conclusion was that some combination of the two schemes might be discovered which would give satisfactory results without the objectionable large pressure drop or the excessive size of the volume. A series of runs was made, while the various volumes were in the line, where orifices of various sizes were placed

at the entrance and exit of the volumes. The venturi and the orifice meters were used in these tests. It was found that it was possible with a volume of several cubic feet, combined with a pressure drop of about two inches of mercury, to reduce the error to a small figure, even for a meter having a large maximum error.

79 *The Muffler as a Quieting Device.* Following the experiments with the volumes and orifices combined as a means of eliminating the pulsation, the idea was further developed by the combination of a volume with several orifices; or in other words,

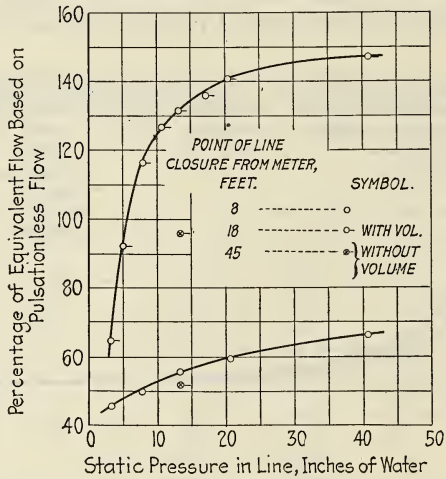

Fig. 27 Effect of Pulsation on Venturi Meter in "Dead-End" Line

the adaptation of the principle of the automobile muffler to the problem of pulsating flow.

80 To study the effects of a muffler a device was constructed of 8-in. pipe-flange sections, as in *B*, Fig. 8. The curve given in Fig. 26 is based upon all the results obtained for every type of muffler tested. The results for the 8-section muffler were more complete and show at the upper limit an error of 81.5 per cent for pulsating flow for open pipe. The whole curve shows that the effectiveness of any single type of muffler, aside from its value as a volume alone, depends entirely upon the amount of throttling produced and very little upon the design and arrangement of its baffle work.

POSSIBILITY OF ADJUSTMENT OF ERRORS

81 There is a possibility that the error shown by a meter when measuring pulsating flow may be so adjusted or so compensated for, that the true quantity passing through the meter may be known. This is mentioned as a possibility only, and forms a basis for comments on some of the circumstances which would attend such an attempt.

82 There is the first possibility of the integration of the curve, obtained by means of the photopulsometer, which is a series of

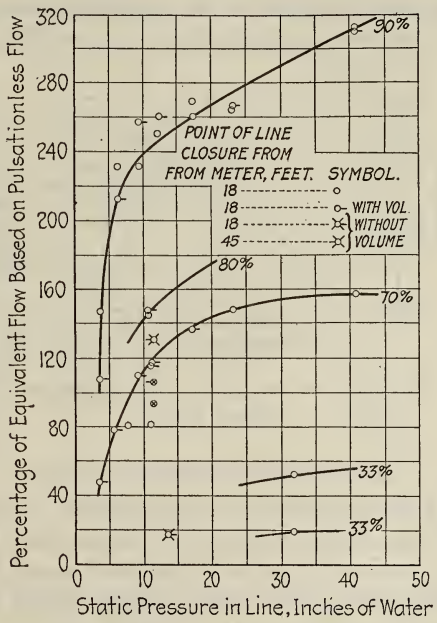

Fig. 28 Effect of Pulsation on Orifice Meter in "Dead-End" Line

velocity-time diagrams. The second possibility would be the use of a correction factor for the manometer itself. A third possibility would be the determination of a pulsating factor by calibration under actual running conditions.

83 *Effect Produced by "Dead-End" Pulsation.* During our experiments in connection with the effect of pulsations on the static pressure we had noticed indications of a reading on the meter for zero velocity in the line. Later a series of tests was made with each of the four meters in while the test line was closed at different points

beyond the meter station. The static pressure, at a point in the line 6 ft. above the meter, was maintained the same as during the regular line of tests for 0.9 in. static head at the orifice head. This static pressure was secured by regulating a by-pass valve in the line 18 ft. above the meter (see Fig. 10). The line could also be closed at flanges located at 8 ft., 18 ft., 40 ft., and at the orifice head, 45 ft. below the meter. For the "dead-end" tests most of the runs were made with the large tank, *T*, in the line, and a few runs were made with the large tank out of the line.

84 The effect produced by the "dead-end" pulsation is represented by the flow equivalent to the reading on the meter as compared with the flow under pulsationless-flow conditions. The curves

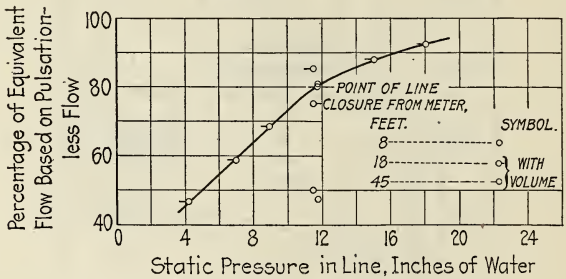

FIG. 29 EFFECT OF PULSATION ON 1½-IN. FLANGE-NOZZLE METER IN "DEAD-END" LINE

shown in Figs. 27–31 show this relation for the four types of meters. The standard flow conditions as represented, during the regular runs, by the 0.9 in. static head at the orifice head, the static pressure in the line above the meter would read about 9.5 in. of water for pulsationless flow and the corresponding static head under pulsating flow would range from 9.5 in. to 16 in. of water, according to the type of meter used. Generally speaking, as would be expected, the static pressure in the line read a little higher under pulsating flow than under pulsationless flow, in some cases showing an increase of 3.5 in. in 10 in. This would be equal to an error of 16 per cent due to pulsation.

85 The curves, Figs. 27–31, give, especially, the effect due to change in static pressure in the line while the meters are in a "dead-end" line. For all meters there is an increase in pulsating effect as the static end is increased up to 20 in., after which they increase more gradually up to 40 in. static pressure, the limit of the tests.

86 In most cases there was a tendency toward an increase in the false flow reading with increase of distance to point of line closure from the meter. In two instances the meter showed a negative reading. The rapid change in this effect on the meter with variation in static pressure, together with the marked variation in apparent flow with change in point of line closure from the meter, makes it exceedingly doubtful whether any reliability could be placed on this method of obtaining the pulsation factor for any given installation.

87 *Application of Proposed Formula to Pulsating Flow.* An attempt has been made to test the relation expressed by the follow-

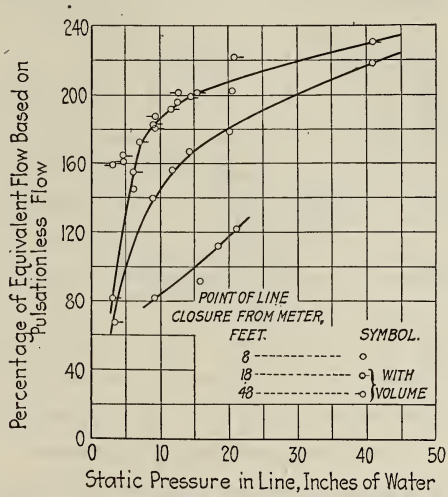

Fig. 30 Effect of Pulsation on Pitot Tube No. 1 in "Dead-End" Line

ing formula by means of the "dead-end" flow data. This formula[1] is: $P_2/(\sqrt{P_1} \pm \sqrt{D})^2 \times 100 = 100$ per cent, where P_1, P_2 and D are respectively the corrected manometer readings for pulsationless flow, pulsating flow, and "dead-end" flow. The positive value of D has been used since it was considered to be better adapted to the data taken.

88 Fig. 32 shows the relation between the ratio P_2 to $(\sqrt{P_1} \pm \sqrt{D})^2$ and the point of line closure. The curve in broken lines is the average for the total number of results; the full line is the average for the few points for the line without the volume, Q. For the 8-ft. distance the results are widely scattered and indicate

[1] Measurement of Gas and Liquids by Orifice Meter, H. P. Westcott, 1922.

that this distance is too close to the meter. The 18-ft. distance gives results showing more uniformity with the average close to 100 per cent. This would indicate the formula would apply better for line closure at 18 ft. The 45-ft. point, though representing the end of the duplicate line, with an average ratio falling below 100 per cent, would indicate less agreement with the formula. The volume Q does not make any marked effect, or as much effect as is shown by difference in length of line closure from the meter.

89 The uncertainty as to where the point of closure should be made or as to whether a length of line duplicating any given

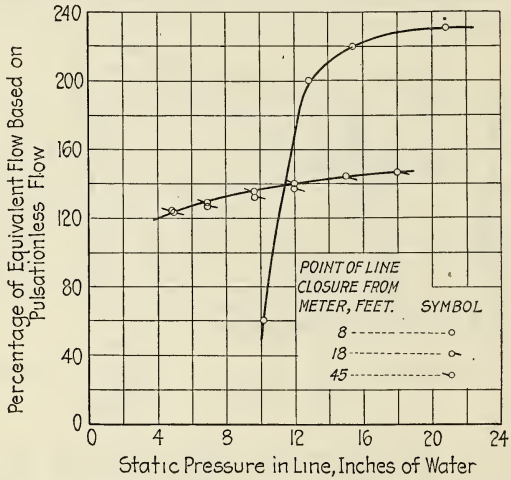

Fig. 31 Effect of Pulsation on Pitot Tube No. 2 in "Dead-End" Line

line would give the true pulsating-head factor make it doubtful whether the proposed formula could be applied with any assurance of a reasonable degree of accuracy.

CONCLUSIONS

90 This investigation relating to our special installation is summarized as follows:

A Nature of Pulsations:

a Pulsations in a pipe line, originating from a reciprocating piston, or a similarly disturbing system, consist of sudden changes both in the velocity and in the pressure of the fluid.

b The pressure change is the most apparent and is probably the greatest factor in producing errors in metering devices.

c The pressure change is in the form of a wave front resembling a traveling sound wave of low frequency.

d The pressure wave travels in the pipe with the velocity of sound.

e The velocity of the pulsation is independent of the velocity, or quantity, of fluid flowing.

f Pulsations in air flow are similar to the compression waves set up by water hammer. Both travel at the velocity of sound in the fluid and are independent of the velocity of flow.

g The effect of this pulsation on a flow meter is to increase its reading, often causing an error of great magnitude. The magnitude

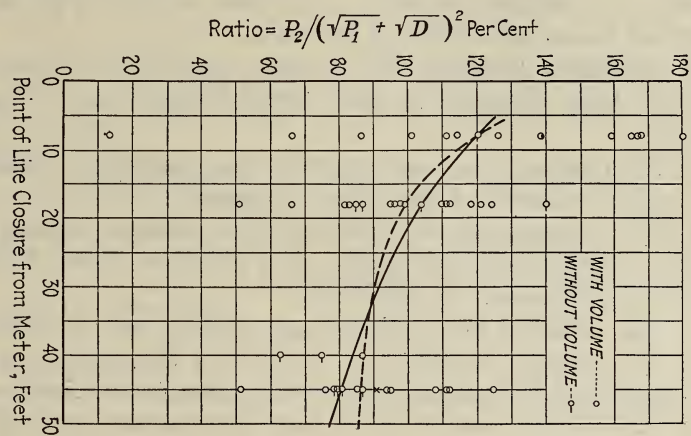

FIG. 32 EFFECT OF LINE CLOSURE ON PROPOSED FORMULA — WITH "DEAD-END" PULSATIONS

of this error depends upon the frequency of pulsation, nominal static pressure of the fluid, type of meter used and adjacent fixtures in the pipe line.

h With orifice meters and flange nozzle meters the pulsating error increases as the diameter of the orifice, or nozzle, approaches the diameter of the pipe.

i The throttling or modification of the manometer connections to the meter does not appreciably reduce the error.

j The point of attachment of manometer connection has no great effect on the error due to pulsating flow.

k The pulsation error at the center of the pipe is 35 per cent less than that at the wall of the pipe.

l A meter on a "dead-end" connection will usually show a positive error of considerable magnitude.

m The pulsation must be eliminated or greatly reduced in order to have the meter read without objectionable error.

B Practical Elimination of Pulsations:

n Because of the high velocity of the pulsation, an excessive length of pipe line would be necessary to destroy the pulsation.

o Throttling is effective but requires a pressure drop of 6 in. of mercury to reduce the error to 5 per cent.

p Abrupt volume enlargements in the pipe line will eliminate the error, if of sufficient capacity. A volume capacity of 20 cu. ft. is required for an error within 2 per cent.

q Generally speaking, for the same capacity, a volume of relatively large diameter is more effective than one of small diameter.

r No relation was found between the compressor displacement and the capacity of the volume chambers.

s The combination of throttling with volumes forming the "muffler" device probably is the most effective device for the mechanical elimination of pulsations.

t The pulsating bag, or diaphragm, and the fan, or revolving baffles, are partially successful in eliminating the pulsations, but their installation is thought to offer serious practical objections.

u The effectiveness of any of these quieting devices seems to depend upon their ability to dissipate or change the energy of pulsation which is effected chiefly through a drop in pressure.

v The device which will destroy the pulsating energy with the least obstruction to the flow of the fluid is the most desirable.

w The effectiveness of the meter element itself in quieting the pulsation depends upon the degree of restoration of the pressure beyond the meter. The greater the percentage of restoration, the higher the percentage of error shown for any given type of meter.

C Adjustment of Error of Pulsation:

x It is probably not feasible to correct any meter by means of a correction factor owing to the disturbing effects which may arise from slight changes in the installation and running conditions.

y The experimental establishment of a pulsating correction

factor and its relation as shown in the proposed formula is not considered feasible with our present experimental knowledge of the laws of pulsating flow.

z It seems probable that each installation where pulsating flow is present would present its own peculiar problem for which an individual study and consideration of the existing conditions would be necessary for a satisfactory solution.

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DISCUSSION

H. N. Packard. We do not agree that the "pressure change . . . is the greatest factor in producing errors in metering devices." For instance, imagine a compressor cylinder discharging through a short length of pipe, in which is mounted a pitot tube, into a large volume such as a gasometer. In the pipe section no measurable static pressure change during the cylinder discharge can be detected, but a very appreciable variation in rate of flow must occur with the consequent error of meter reading. This is readily confirmed by actual test with the pitot tube. As a further proof of this point, in the Thomas meter, which is entirely independent of pressure conditions and indicates only the standard units flowing through it, we have errors with heavily pulsating flows. Assuming a pulsating flow of sine wave form, the meter error is negligible up to the point where the flow at the peak of the wave does not vary more than 30 per cent from the mean rate. When the wave amplitude is as great as the mean flow, giving instantaneous stoppage of flow in each cycle, the meter error reaches a maximum of nearly 8 per cent. As practically all meter installations are fairly close to the pulsation producing piston, we believe the errors are mostly due to actual instantaneous flow variations through the metering device.

We believe that the velocity of transmission of the wave is that of sound plus that of the gas velocity, slightly modified by the size and shape of the pipe line. We have found a number of references in standard works on physics that the velocity of sound in air is dependent on the wind velocity as well as the density of the air. Do the authors feel that their data can support their state-

ment? Their maximum test velocity was 27 ft. per sec. compared with about 1100 ft. per sec. for sound, and it seemed to us difficult to be sure that the experimental error was not as great as this.

The curves on Fig. 23 would indicate some reduction in manometer error with throttling of connections. Have the authors any explanation of this other than possible leaks? We can see no other reason for the gain in accuracy.

The statement is made that at least a 6-in. mercury pressure drop is required to reduce pulsations to a practical limit. Do the authors consider this a general statement or applicable only to their test conditions? It would appear to me to be a function of the density of the fluid, its velocity and the pulsation wave form (magnitude of pulsation) if made as a general statement.

We are still of the opinion that there is some relation between the piston displacement and volume of a quieting receiver which will give good results. Taking the two absurd extremes of a volume equal to piston displacement and an infinite volume, in one case we know that no effect will be produced and in the other perfect quieting of pulsations will occur. We believe that the quantity of fluid discharged per stroke, the number of strokes per minute and the volume and diameter between the source of pulsations and the meter determine the pulsation effect at the meter, at least with elastic media such as gases.

On dead-end error tests we believe that there is an actual displacement of gas back and forth in the meter, this flow effect being due to the elasticity of the gas which is alternately compressed and expanded in the dead-end volume. In our own meter the dead-end effect does not occur at all except with such large dead-end volume that the alternate gas flows through it, traveling at least seven inches in a reverse direction. This condition has been met but once in our commercial experience. Our meters are normally set vertically with the normal flow vertically upward. At zero load a very small amount of heat is still left in the heater to maintain control and due to convection current the heated gas rises through the exit thermometer and maintains its temperature higher than the entrance thermometer sufficiently to give a constant tendency to shut down the meter. With large dead-end volume and severe pulsations there is sufficient back flow to overcome the convection currents and carry heated gas from the heater into the entrance thermometer, increasing its temperature to approximately the same as the exit thermometer. Immediately the control for the meter goes to full load under such conditions.

J. M. Spitzglass. Prior to the advance of Professor Judd's experimental work on pulsating flow there was an idea prevalent that the error in the measurement was due mainly to the magnifying effect of the differential column, reading the average height and the corresponding square root of this average instead of the average of the instantaneous square roots which are the equivalent of the varying flow.

With the development of the flow meter, we sought to eliminate this error by making the meter respond electrically to the instantaneous instead of the average height of the differential column. This provision was thought to eliminate the part of the error which the authors of the paper designate as the "effect of the type of manometer used." We soon discovered that there was a much larger error due to the "harmonic" effect of the pulsations in the flow. Still, in all our observations with reciprocating flow this error seldom exceeded 25 per cent under any circumstances. Furthermore, this error could be easily eliminated by moderate restrictions in the form of additional orifice plates on either side of the differential medium.

The writer was greatly surprised when he first glanced at the tables in the paper to note that the errors in some of the meters were as high as 200 per cent and over. The writer does not for a moment cast any doubt on the data of the given observations, but he has felt from the beginning that there must be something misleading in the algebraic presentation of the results. After reading the paper a second and third time, the solution of the riddle presented itself very clearly.

In summarizing the tabular results of the investigation, the authors adopted the velocity pressure of the flow as *the* basis for comparing the effect of the pressure pulsations, which, according to their own explanations and results, was not in the least a function of that velocity pressure. What they actually did was to compare a variable quantity, the pressure pulsations, on the basis of another and more variable quantity, the velocity pressure of the flow in the given meter. It will be observed, therefore, that whenever the assumed basis, the velocity pressure of the meter, decreased, the apparent percentage of error increased in a corresponding ratio.

The writer believes that this is the real reason why the percentage of error increased when the size of the orifice or the flange nozzle was increased. The same reasoning applies and is a sufficient explanation for the fact that the percentage error increased in the

case of the pitot tube at the wall of the pipe, where the velocity pressure (the assumed basis) is the lowest. Furthermore, the dead-end phenomenon shows conclusively that the effect of pulsation is not a factor of the velocity pressure of the air.

As the data and the results of the investigation are exceedingly important for the users of the meter on pulsating flow, the writer would like to ask the authors to include in the paper a summary table, similar to Table 4, giving the actual values of the static pressure in the line, the actual velocity pressure, and the difference in static pressure between the pulsating and pulsationless flow for all meters tested. It can be readily seen that if the difference has a value of, say, 5-in. of water, it may form a square-root error of 145 per cent on a meter whose velocity pressure is only one inch of water, while the same 5-in. difference will form an error of only 5 per cent on a meter whose velocity pressure amounts to 50-in. of water; and what is more important, if by means of moderate restriction or increased volume the difference is reduced from 5 to 0.5 in., it will still give an error of 22 per cent in the first case and will be entirely insignificant in the second case. To state this in another way: The effect of pulsation, according to the writer's understanding of the investigation, is shown to be rather in the nature of an additional term than a factor in the algebraic expression of the flow for a given meter.

R. J. S. Pigott. The paper on pulsating flow reports the results of research work undertaken for the Special Committee on Fluid Meters, of the Society's Research Committee.

One point about pulsating flow seems to be coming more strongly to the fore; that is, the problem is largely an acoustic one. All the data go to show that the variability of the conditions is due to the fact that the acoustic conditions in the pipe differ with every installation, and it is hard to see how pulsating flow can be stopped in every case until a study is made of the phenomena from an acoustic standpoint.

Up to the present time we have not developed devices for detecting the acoustic variations. We have been working along lines of mechanical devices almost wholly, and have never given enough attention, as yet, to demonstrating clearly what are the acoustic conditions in the pipes.

One of the earliest problems in the opinion of the members of the Fluid Meters Committee was that of either providing correction

for the effect of pulsating flow upon the indications yielded by the flow meter mechanism, or to so reduce the pulsations as to render their effect insignificant. The net result of research has definitely confirmed the belief that any type of meter operating on a difference of head which is proportioned to the square of velocity will register high on pulsating flow. The other belief, that it is very difficult, if not impossible, to provide suitable correction factors for the readings of the meter, is also very largely confirmed. The problem, therefore, is mainly reduced to providing commercially practicable means for suppressing pulsations to a point where they do not have a marked effect in the registration of the meter.

The experimental work so far carried out has indicated that it is feasible to accomplish this end by means of a combination of throttling with enlargement of volume. The scope of the experimental work has not been large enough as yet to definitely establish the amount of throttling and the amount of volume enlargement required for any particular case and it is probable that the variations in velocity and size of lines will render an exact solution for any specific case difficult. However, there is a good deal of encouragement to be drawn from the evidence that in all likelihood proper relations of the two requirements can be determined to cover a considerable variation in pulsation condition. The situation on pulsating flow is only one factor of several which have tended to reduce the reliance placed upon flow meters for accurate measurements, and it is not generally recognized that the inaccuracies found in the commercial operation of flow meters are due almost entirely to the conditions under which the flow meter is installed. Under proper installation conditions, it can be demonstrated that any well designed flow meter can give results with perfectly satisfactory precision. However, in the great majority of installations insufficient attention is given to the effect of pulsating flow, eddy currents in the lines due to valves, elbows or similar obstructions and leakage in the lines, transmitting a pressure difference to the meter indicating and recording mechanism.

The first report of the Fluid Meters Committee was to have been produced for the Annual Meeting but the amount of work to be done both in editing the report and preparing for printing was too much to permit publication at this time.

This report will cover the matter of installation very fully as well as the theory and accuracy of the devices employed. It is to be hoped that this report will provide for the designers and users

of flow meters a complete summary of information available on the subject. Hitherto there has been no single source from which this information could be obtained and it has been scattered through three or four hundred different publications.

JOHN L. HODGSON.[1] In the opinion of the writer, the results obtained from the elaborate researches described in the paper might have been very much greater had a careful analysis been made beforehand of the ways in which pulsating flows cause errors in meters which are based upon the measurement of a differential pressure.

By making such an analysis the writer has found it possible to reach wider and more general conclusions than the authors of the present paper at the expense of far less experimental work.

Some of the most important of these conclusions are summarized below.

A pulsating air flow may be considered to consist of:

a A "pressure variation" which is transmitted with the velocity of the fluid in the pipe, plus the velocity of the sound in the fluid proper to the particular size and roughness of pipe used, and the nearness to the source of pulsation of the point where the velocity is measured.

b A "velocity variation" during which the whole of the air in the pipe is accelerated or retarded.

The fluid at a point distant from the source of pulsation does not however change its velocity until the impulse, transmitted with the velocity stated under *a* reaches it.[2]

Both these pressure and velocity variations cause errors[3] in the meter; but in quite different ways.

The error due to the pressure variation occurs when the pressure pipes leading to the meter have different coefficients of discharge for inflows and outflows, and when the capacity in the meter on the two sides of the water or mercury column are different.[4] It is then

[1] Eggington House, Beds, England.

[2] The authors state that the velocity of propagation of the impulse is that of sound. That may *apparently* be so in the case of their particular experiments; as the effect of the pipe walls is to retard the speed, and of the moving air to increase it.

[3] The above sources of error, and also the effects of "square root" and "viscous" damping of the manometer or meter were pointed out in a paper by the writer in 1916; see *Proc.* Inst. C. E., vol. CCIV, p. 134 to 137.

[4] In the case of the photo-pulsometer, shown in Fig. 11, the upstream

possible to obtain an actual difference of pressure on the two sides of the meter by the pressure variation alone and when there is no velocity variation at all in the pipe.[1]

The error due to the pressure variation[2] may easily be brought down to a very small amount by using pressure connections which have equal coefficients of discharge in both directions, and by keeping the capacities in the meter about equal.

There remains the error due to velocity variation, which is the real source of trouble.

This causes error because the flow depends upon the mean of the square root of the pressure differences across the measuring device; whereas the meter reading[3] depends (approximately)[4] upon the mean of the pressure differences.

It can be shown by calculation that for certain wave forms this "velocity variation" may produce errors of several hundred per cent.

The error due to this cause can be calculated or determined by calibration for any particular conditions; but as it varies with the rate of flow, and the wave form, and the product of the specific volume and the absolute pressure of the fluid, and the loss of pressure in, and the capacity of, the pipe line, it is best reduced to a small amount rather than allowed for.

The only way to reduce it is to smooth out the wave form of the "velocity variation" at the metering point.

This can be done in many ways, the simplest of which (not mentioned by the authors) is to insert a capacity *and* a throttling

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cavity is larger than the downstream cavity, with the result that when a change of pressure occurs the pressure will rise and fall most quickly in the downstream cavity; thus exaggerating all the readings. Many of the diagrams taken with this instrument indicate negative flows (instead of the positive flows which must actually have existed) because of this defect.

[1] The "dead end" condition referred to by the authors.

[2] The "pressure change" is wrongly stated by the authors to be "probably the greatest factor in producing errors." The "pressure change" is more rightly considered as a "symptom" than a "cause."

[3] That is, assuming that the meter is adjusted so as to show no appreciable error due to the "pressure variation."

[4] This is only true when the "damping" in the pressure pipes and the meter follows the "viscous" law. If it follows the "square-root" law the mean meter reading is not the true mean of the pressure differences. This may explain the change of manometer error as the pressure pipes are throttled shown in the authors' Fig. 23. The data given in the paper are, however, insufficient to enable the point to be settled.

device between the source of pulsation and the metering point. If the meter itself offers sufficient resistance it may form the throttling device; if it does not, an additional throttling device may be added. The capacity should be placed between the source of pulsation and the meter, and the additional throttling device, if any, should be placed on the downstream side of the meter. (See Fig. 33.).

In a paper read before the Midland Institute of Mining, Civil and Mechanical Engineers in January 1921, and again in a paper read before the Institute of Naval Architects in April 1922, the writer showed that, if certain assumptions were made in order to simplify the reasoning [1], it could be proved [2] that the percentage error of a meter for any particular wave form depended upon the value of the ratio:

$$FCL/ZQ$$

where F is the frequency with which the wave form repeats itself

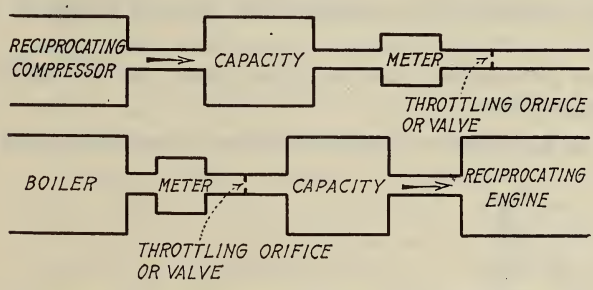

Fig. 33 Location of Capacity with Respect to Source of Pulsation and Meter

C is the capacity, and

L is the loss of pressure between the entrance to the capacity C and the side of the metering device which is furthest from the source of pulsation

Z is the product of the specific volume of the fluid and its absolute pressure

Q is the rate of weight flow.

[1] Such as that the meter is adjusted so that the "pressure variation" causes no error; that the only error is caused by the meter taking the mean of the pressure differences across the measuring device, instead of the mean of the square root of the pressure differences; that L varies in Q^2, etc.

[2] For an elementary proof, see the writer's paper published in the *Proc.*, Inst. Naval Architects, April 1922.

The writer has developed methods which enable curves connecting the values of FCL/ZQ and the per cent error of the meter due to the "velocity variation" to be calculated and drawn out, so that, given the wave form, and the values of FZQ, the value of CL

FIG. 34 SQUARE WAVE FORM

which will reduce the pulsation error to small limits can be immediately read off.

Such a curve for a square wave form, Fig. 34, is shown in Fig. 35.[1]

Many interesting results follow from the FCL/ZQ relation, among which are:

1 A large meter error may often be reduced to a negligible

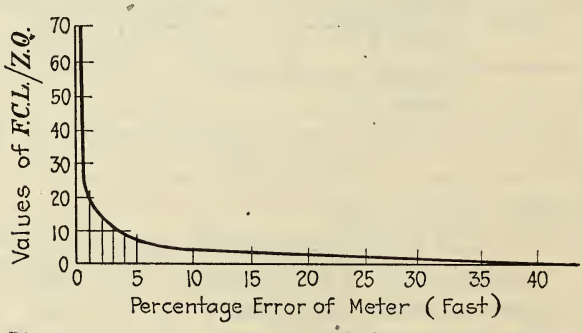

FIG. 35 RELATION OF VALUES OF FCL/ZQ AND PER CENT OF ERROR OF METER

amount by trebling or quadrupling the value of CL by providing for additional throttling and by putting the meter further from the source of pulsation so as to secure additional capacity. It will be seen that in order to reduce the pulsation error, it is equally efficacious to

[1] Compare the authors' Fig. 26, which is plotted against L only.

increase C or L. The energy lost is, however, least when the pulsation is reduced by increasing C.

2 If the rate of flow is reduced on account of the compressor slowing down (the wave form remaining the same) the meter error will be increased, as F and L are reduced, and L falls off more rapidly than Q.

3 Similarly when a steam engine governs, the meter error increases as the flow is reduced. The increase in the error is greatest when the engine governs on the "cut off" (instead of "on the throttle"), as the wave form is then altered for the worse. If there is an appreciable steam chest capacity on the engine side of the throttle, the pulsation error may actually be reduced when the engine governs "on the throttle."

4 If the throttling orifice shown in Fig. 33 consists of a valve (instead of a fixed orifice) and this valve is shut down (either automatically or by hand) as the flow is reduced so that the original L is maintained, the pulsation error will remain constant at all flows, if F and Q fall off in the same ratio.

5 It will be seen on reference to Fig. 35 that if the pulsation error is large, any small change in the value of FCL/ZQ will cause a large change in the pulsation error, whereas if the pulsation error is reduced to 1 or 2 per cent there may be large variations in the value of FCL/ZQ without causing any appreciable change in the overall error of the meter. It is therefore far better to reduce the pulsation error to a small amount by increasing FCL/ZQ than it is to "rate" the meter by actual calibration for a large pulsation error.[1]

6 The FCL/ZQ relation explains the authors' conclusions $A(h)$ and $B(w)$, since it shows that the pulsation error is large when L is small.

7 It can be shown that for given values of FLZ and Q the amount that the meter reads fast is (roughly) inversely proportional to C^2, or for given values of CLZ and Q to F^2 and so on.

It should be understood that the FCL/ZQ relation, being deduced from premises which simplify the actual conditions, does

[1] Compare the authors' conclusions $A(m)$ and $C(x)$.

not hold with absolute rigidity in practice, and also that it only holds over a limited range of conditions.[1] At the same time it serves as a most valuable key to the meaning of what is otherwise a mass of unrelated data.

The writer would say that he disagrees with the authors' conclusions $A(b)(d)(e)(f)(i)$, and $B(n)$.

It will be seen that he is generally in agreement with conclusions $C(x)$ and $C(z)$. With regard to the latter he would say that curves connecting the values of CFL/ZQ and the percentage error of the meter which he has had calculated out enable his firm (Messrs. Geo. Kent of London and Luton, England) to decide on the proper value of CL for any particular case at the expenditure of a few minutes' work only.

In conclusion he would like to congratulate the authors upon the scope of their work, and upon the clear way in which they have set forth their results.

The Authors. In reply to Mr. Packard, we can readily see that his type of meter would not be greatly affected, if any, by the pressure changes, even though the static pressure gage might read higher due to the pulsation. We would also conclude from our investigation that his meter would be less affected by the pulsation because the effect on the velocity head seems to show much less error than that produced in meters depending on the pressure drop readings.

In regard to the effect produced by the static pulsation we have failed to convey the proper meaning. The change or effect on static pressure produced by the pulsation is much greater than the effect produced on the velocity head. This pulsation, like sound, seems to be propagated as a pressure wave and the effect produced on any measuring device, especially where difference in pressure head is used, is much greater than the effect recorded on the velocity diagrams from the photo-pulsometer. Hence, the conclusion that the "pressure change" was the greatest disturbing factor was drawn. This we believe to be borne out in our work. By the photo-pulsometer the diagrams with air flow through the venturi showed a maximum velocity change of $1\frac{1}{2}$ in. = 4.5 in. water as taken with the pitot tips which from their construction would produce a magnified reading while for the same flow con-

[1] For instance, it does not hold when the rate of flow is so great that the loss of pressure, L, no longer follows the square root law.

ditions, the indicator diagrams giving static pressure fluctuations showed a change about 1.2 lb. or say 35-in. water or about eight times as great as was shown in the velocity head reading neglecting the fact that the instrument gave a magnified reading.

Furthermore in Fig. 20, the diagrams 1 and 2, section B, show a pressure card and a pressure and velocity head card respectively. A comparison of the two cards show little or no difference and at least indicate no great change, if any, due to the velocity head when added to the static head, as taken by a pitot tip attached to the indicator cock.

It seems apparent, therefore, that the pulsation, (assuming its propagation as a pressure wave) is transmitted in the pipe by means of the air (either flowing or quiet) as a medium; and that with the dead-end meter connection the pulsation is surging back and forth independent of the air which itself may also have some slight movement back and forth. This transmission of pulsation in the dead-end line would seem to be similar in this respect to the surge of pressure in a water line due to water hammer which is very much greater in magnitude as compared with the effect due to velocity.

In regard to the velocity of the pulsations, the statement should be modified to show that the velocity of the pulsation included the velocity of the wave and the velocity of the air. For the maximum velocity of 27 ft. per sec. the error would be less than 3 per cent if we neglect the velocity of the air which of course is much less than the actual experimental error.

The conclusions given in the summary are made with reference to the installation which we tested; also reference to the throttling effects of a six-inch mercury pressure drop considers our test conditions only, and should be modified much according to Mr. Packard's suggestion.

In regard to error due to throttling manometer connections, we do not believe that there were any leaks, since these manometer throttling plugs were made up in sets all of the same material and in the same way, and furthermore the data seemed to be consistent.

Referring to the relation to piston displacement of the volume of a quieting cylinder, it is probably true that *some relation* exists, but it seemed to us that it would take such an extended investigation to establish anything approaching a law, as to render the solution impracticable.

In the dead-end meter installation, we agree with Mr. Packard that there is an actual forward and back flow of the fluid due to

the elasticity of the gas; but it is also true, we think, that the pulsation in the form of the compression wave travels forward in undiminished amplitude and returns in more or less diminished amplitude depending on the length, shape, and volume of the dead-end connection.

Mr. Spitzglass states in his discussion that the authors in finding the per cent of error due to pulsating flow have compared "a variable quantity, the pressure pulsations, on the basis of another and more variable quantity, the velocity pressure of the meter."

The error due to pulsating flow was based on the velocity, or quantity of flow, or its proportional equivalent the square root of the pressure difference through the meter element for pulsationless flow. For the four types of meters used the velocity head is equal to or proportional to the drop, or pressure difference, through the meter element. From whatever cause the pulsating flow may have been produced it is quite evident that its effect would have to be determined from the reading on the meter manometer.

It appears to us, therefore, that while the pressure pulsation seems to be the greatest factor in the error due to pulsating flow it is the velocity head reading that is observed on the meter. In our opinion it is the velocity-head readings as shown by the meter for both conditions of flow that should be compared. In fact we are at a loss to know of any other way of establishing the per cent of error.

Mr. Spitzglass also calls attention to the fact that for the orifice meter and flange nozzle meter the percentages of error increase approximately as the inverse ratio of the velocity heads. The reason he assigns is that as the velocity through the meter element decreases the effect due to the static pressure pulsations increases; and likewise by the same reasoning the apparent increase in percentage of error, as the walls of the pipe are reached, can be explained. This relation for the orifice and flange-nozzle meters can be noted in the following table but it is felt that the data is not sufficient in relation to the behavior of pressure pulsation at different points of the line to warrant more than a mention of this apparent relation.

Owing to the limited capacity of the air compressor it was not possible to experiment with pulsating air flow under varying static pressure conditions. For the dead-end meter connection static pressure changes could be noted and it was observed that a rapid increase in percentage of error occurred with increase of static pressure. Hence in the discussion, where it is stated that a 5-in.

differential would give 145 per cent error for a flow condition of one inch true differential and the same differential of 5 in. would give only 5 per cent error for a true differential of 50 in. of water, it would be unsafe to predict that the differential would still be 5 inches for a 50-in. true differential. A flow under 50-in. head would necessarily require a much higher static pressure which is

TABLE 11 FLOW CONDITIONS IN THE TEST LINE

Meter used	Static pressure in line above meter, in. water			Velocity head by meter man., in. water	Maximum error, Per cent
	Pulseless	Pulsating	Difference		
1	2	3	4	5	6
Venturi	10			9.29	
		13.5	3.5	30.15	81.0
Orifice	27.7			22.00	
33%		32.0	4.3	26.35	9.5
Orifice	11.8			4.10	
50%		13.4	1.6	10.48	60.0
Orifice	9.8			0.94	
70%		11.4	1.6	4.83	128.0
Orifice	9.4			0.41	
80%		10.8	1.4	2.93	167.0
Orifice	8.9			0.0865	
90%		9.6	0.7	1.083	254.0
Flange-nozzle	13.0			5.59	
1⅛ in.		16.3	3.3	11.73	45.0
Flange-nozzle	10.0			1.64	
1½ in.		11.4	1.4	6.16	94.0
Flange-nozzle	9.3			0.47	
2 in.		9.9	0.6	2.74	143.0
Flange-nozzle	8.7			0.121	
2½ in.		9.4	0.7	1.090	200.0
Pitot	8.8			0.163	
No. 1		9.2	0.4	1.110	162.0
Pitot	8.8			0.224	
No. 2		9.6	0.8	1.000	115.0

likely to result in a differential between pulsationless and pulsating flow much greater than the assumed 5 inches and hence the percentage of error would be correspondingly increased.

The authors are glad to comply with Mr. Spitzglass' request to include a table giving in summary a statement of the flow conditions in the test line:

Columns 2 and 3 give the static pressure in the line for pulseless and pulsating flow respectively. Column 4 gives the difference

in static pressure for pulsating flow over pulseless flow for a velocity of flow in the three-inch line of twenty-two feet per second.

It will be noticed that although the flow conditions were uniform for the different meter elements the pulsating effect on the static manometer varied, ranging from 0.4 to 4.3 in. of water. This variation depended upon the kind of meter element used. Those elements which obstructed the pipe most, namely: the venturi, the 33-per cent orifice and the $1\frac{1}{8}$-in. flange-nozzle, showed the greatest effect due to the static pulsation. Where the pipe was least obstructed as with the orifice, flange-nozzle, and pitot meters the static manometer showed the least effect. Also the maximum percentage of error is seen to vary approximately inversely as the error shown by the static manometer.

As pointed out in the paper and as further emphasized by Mr. Pigott, the authors believe that very little can be done to establish suitable correction factors for meters operating under pulsating flow and that the solution of the problem is reached when some suitable means is provided which will reduce the pulsations to a negligible point. The adaptation of the "muffler" device is apparently the most effective mechanical device for reducing the pulsations. However, further study and experimentation is necessary to establish the proper combination of throttling and volume space for static pressures and pulsating conditions approaching those in general practice.

The authors feel greatly honored in having their paper reviewed by Mr. John L. Hodgson of England.

Mr. Hodgson takes exception to certain conclusions in the paper, in some cases justly so, and in others due apparently to a wrong interpretation of the paper. He points out the importance of having equal spaces in the manometer connections of the meter and in the case of the photo-pulsometer equal spaces above and below the diaphragm. The authors also recognized the importance of this and so far as possible all manometer connections were made of equal length, although this relation could not be maintained while the manometers were in use. Our experiments showed (see Fig. 23) that, when the manometer tubes were throttled, a throttle-plug, with a diameter of 0.02 in. was necessary to reduce the error from 80 to 60 per cent for the venturi meter, with throttling plug coefficients the same for flow in either direction and other conditions of flow remaining constant. The use of volumes in the manometer connections in various combinations and with uniform and similar

connecting tubes, $\frac{1}{8}$ in. diameter opening (see "M," Fig. 4), gave no more favorable results than did the throttle plugs.

With the photo-pulsometer the space below the diaphragm was made equal to the space above as were also the connections to the searching tubes as shown in the corrected drawing of Fig. 11. A partition with a glass shutter extended across the diaphragm chamber so that the diaphragm was equidistant from each wall. The diagrams showing negative pressure could not therefore be caused by unequal capacities. The photo-pulsometer was used chiefly as an indicator for illustrating and confirming pulsationless flow conditions.

It is conceivable that the pressure pulsations might be lessened, perhaps nearly eliminated, by the use of the proper amount of throttling, observing at the same time that equal spaces were provided at the manometer connections for varying quantities of flow. We concluded that it was an extremely doubtful and uncertain method, if not wholly dangerous, to rely too fully on such expedients for reducing the pulsating error.

The question raised in regard to the relative effect of the pulsation on the static pressure is answered, we believe, in the reply to Mr. Packard. The authors have designated as static pulsation changes all pulsating changes which act in a direction at right angles to the stream flow. For the venturi meter, with the usual static pressure connections, acting under pulsating flow, the liquid in the manometer tubes would steadily rise until a differential head was reached equal to more than three times the differential head due to pulsationless flow. The column would vibrate through a range of $\frac{1}{2}$ to $\frac{3}{4}$ of an inch corresponding in frequency to that of the flowing air. This vibration is due, in our opinion, to the in and out flow through the manometer tubes, and represents the only effect on the manometer tubes due to the velocity pulsation, the greater part of the manometer reading being due to the static pulsations.

We agree with Mr. Hodgson and Mr. Packard that the velocity of propagation of the pulsating wave approaches that of sound in the flowing fluid plus the velocity of the flowing air. However, in our opinion, as based on our experiments, we are not willing to concede that the pressure pulsation has a less effect than the velocity pulsation in meter installations where these pulsations act on manometers with static connections at right angles and even with the inside of the pipe. From the "dead-end" flow experiments it

would also seem evident that the pressure pulsation effect is many times greater than that due to the velocity pulsation.

The statement was made by Mr. Hodgson that the simplest way of reducing the pulsation by the use of a capacity combined with throttling was not mentioned by the authors. In conclusion *B* (*s*) we have stated that the "muffler" device, a combination of capacity, or volume, with throttling is probably "the most effective device for the mechanical elimination of pulsations." The capacity or "muffler" was always inserted between the disturbing element and the meter. It would not seem advisable in our opinion, to insert the throttling device in the line below the meter. It would seem better to eliminate the pulsations as far as possible before the meter was reached.

The authors regret that Mr. Hodgson's paper containing his latest investigations involving the development of a formula for pulsating flow was not available for examination and study until after their paper was written. The grouping of the factors involved in pulsating flow in order shown in the formula seems feasible. Mr. Hodgson is to be congratulated in being able to reduce the results of his researches to a working formula which shows the factors involved in their proper relation. We are in full accord with the opinion of Mr. Hodgson that the only sure way to meter pulsating flow is to reduce the pulsations by means of suitable capacity and throttling, and we likewise believe that even the formula proposed, or any similar formula, while serving in a general way cannot be too rigidly applied in practice. Each installation will present its own peculiar problem.

No. 1870

ORIFICE COEFFICIENTS—DATA AND RESULTS OF TESTS

By Jacob M. Spitzglass, Chicago, Ill.
Member of the Society

This paper discusses the results of experimental work conducted at the Research Laboratory of the Republic Flow Meters Co., of Chicago, on the use of orifice plates for measuring the flow of fluids in pipes.

Extensive tests were carried on for a period of several years to determine the effect of the varying factors such as the orifice ratio, the size of the pipe, and the distance of the upstream and downstream connections upon the pressure difference across the orifice plate. The tests were conducted on seven sizes of pipes, from 2-in. to 12-in. standard. Fourteen orifice ratios were used in each size and twelve connections were tested on each side of the orifice plate.

The tests were made on the flow of atmospheric air, supplied by a Sturtevant blower having a maximum capacity of 6000 cu. ft. per min., at a pressure of 10 in. of water. The air passing through the orifice was measured by five calibrated impact tubes ranging from 1 in. to 4 in. in internal diameter. A detailed account is given of the calibration of the impact tubes, and curves were plotted showing the coefficient as a function of the impact head through the tube.

The arrangement of the installation for the final tests is described and illustrated in detail, and an account is given of the method of testing the various orifices. The theoretical relation of the functions is given briefly, preceded by elaborate definitions of the terms involved. The paper includes a number of tables and a collection of characteristic curves that may be employed for determining the flow through orifice plates in a given size of pipe.

IT HAS been long established in the literature of mechanical engineering, and especially in the various textbooks on hydraulics, that a stream of fluid issuing from a rounded aperture in the wall of a large vessel can be gaged accurately by the difference in static pressure between the inside and the outside of the vessel containing that fluid.

2 The relation between the pressure difference and the velocity

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of the jet issuing from the aperture has been derived theoretically for jets flowing into the open atmosphere and also for jets submerged under a lower head of the same fluid. It has been established by numerous experiments that upon leaving the vessel the cross-section of the stream is gradually reduced in area until, at a point known as the vena contracta, it is only about 60 per cent of the actual area of the aperture in the wall of the vessel. This contrivance used as a device for measuring the flow of fluids has been termed the "frictionless" or "thin-plate" orifice.

3 Recently the principle of the thin-plate orifice has been successfully applied to measuring the flow of fluids in pipes by the drop of pressure through the restricted area of an orifice plate placed between two flanges of the given pipe. At first the practice was limited entirely to cases where the area of the opening was much smaller than the cross-section of pipe, so that they could be considered as being similar to the submerged aperture in the wall of a large vessel. Further developments in this application have necessitated the use of larger-ratio orifices, which obviously differ from the case of the jet issuing from the large vessel, and therefore cannot be treated in the same manner. A comparison of the two cases with reference to Fig. 1 brings out the following points of difference:

4 First, in the case of the thin-plate orifice in the large vessel, the velocity of the jet issuing from the wall represents the equivalent of the hydrostatic head or pressure above the center of the opening, since the fluid in the vessel may be considered to be at rest. In the case of the pipe orifice it is the increase in velocity over the velocity of approach that is equivalent to the differential head across the orifice. This item may be termed the effect of initial velocity in the pipe.

5 Second, in the case of the pipe orifice the formation of the stream may be greatly affected by the proximity of the walls of the pipe both on entering and on leaving the orifice. In the case of the large vessel this effect is eliminated on either side of the orifice and the stream of the fluid can be freely formed into the natural contraction of the issuing jet. This item may be termed the effect of the size of the pipe or of the proximity of the walls.

6 Third, in the case of the large vessel the pressure difference affecting the flow of the fluid is fixed by the difference between the pressure inside and outside the vessel. In the case of the pipe orifice it is necessary to obtain the pressure difference between a

point ahead of the orifice where the current is parallel to the walls of the pipe, and a point beyond it where the jet has been fully contracted, in order to have the true equivalent of the increased velocity pressure of the jet. This may be termed the effect of shifting the pressure taps on either side of the orifice.

7 These given points of difference are demonstrated graphically in Fig. 1 to show their relations to the dimensional quantities involved. Thus, the effect of initial velocity is a function of the relation between the velocity V of the fluid in the pipe and the velocity v of the jet at the point of greatest contraction, or of the ratio d^2/D^2 between the cross-sections, which, in its turn, is a function of the jet contraction X and of the ratio between the cross-

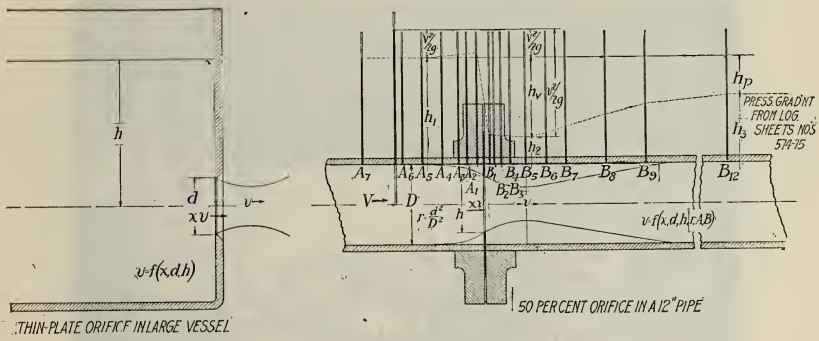


FIG. 1 THIN-PLATE ORIFICE — JET CONTRACTION AND PRESSURE GRADIENT

section of the orifice and the cross-section of the pipe. The effect of restriction or the proximity of the walls is a function of the distance between the wall of the pipe and the edge of the opening, which, in its turn, is a function of the diameter D of the pipe for any given orifice ratio. The effect of shifting the pressure taps is necessarily a function of the location of the taps on the A and B sides of the orifice plane.

8 The experimental work described in the following paragraphs was undertaken with the object of obtaining the necessary data for this analysis: namely, to determine by actual tests the effect produced in the pipe orifice, used as a flow-measuring device, by the variation of the orifice ratio, by changing the size of the pipe, and by shifting the pressure taps on either side of the orifice plate.

PRELIMINARY TESTS

9 In the early part of 1919 the engineering department of the Republic Flow Meters Company began its first series of orifice tests on several sizes of pipes. It was realized at the start that an enormous number of comparative tests would be necessary to take care of all the variables involved. The plan was laid out to test seven sizes of pipes, from 2-in. to 12-in. standard. It was considered necessary to prepare fourteen orifice plates for each size, varying the orifice ratio from 5 to 70 per cent. It was desired to make twenty-four pressure taps on each pipe, twelve on the upstream side and twelve on the downstream side of the orifice plates. Accordingly, the number of tests involved would be seven for size of pipe, fourteen for orifice ratio, twelve for upstream distances, twelve for downstream distances, and, varying one condition at a time, would necessitate at least $7 \times 14 \times 12 \times 12$ or a total of 14,112 tests; and counting duplications and discarded tests, probably a total of 20,000 in all. To make a reliable quantitative test with any kind of fluid — steam, water, air, or gas — it would require at least one hour for each condition at varying velocities, not counting preparations and adjustments. Running constantly at the rate of six tests per day, three hundred days a year, it would require eleven years for the completion of the work. It was obvious, therefore,

Fig. 2 Interior of Research Laboratory

that a simplified method of testing was necessary in order to accomplish the work in a reasonable amount of time.

10 The idea was conceived at that time of laying out a system of testing whereby the variables involved could be determined for large-ratio orifices by comparison with the known or approved and tested value of the coefficient in small-ratio orifices, approaching the condition of the rounded aperture in the large vessel. Immediately a trial installation was made on three sizes of pipes, 2-in., 4-in. and 6-in. standard, and for the basis of comparison two impact nozzles were prepared, one 1 in. in diameter and the other 2 in., and these were calibrated by a 10 per cent orifice in the 4-in. pipe.

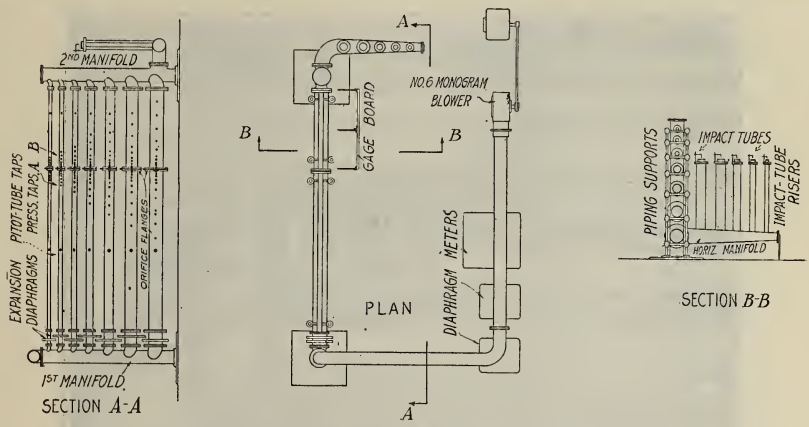


Fig. 3 General Layout of Research Laboratory Equipment

11 The method of procedure was, first, to determine the coefficient of the impact tube by comparing it with the 10 per cent orifice in the 4-in. pipe. The coefficient of the orifice at that ratio was taken arbitrarily to be 0.61, making some allowance for the initial velocity and the proximity of the walls of the pipe. The method of testing involved merely the comparison of coefficients in either case, which could be done by taking a momentary reading of the gages for each test.

12 A number of major tests were performed on that installation and the results were plotted tentatively for comparison. The curves obtained have demonstrated the importance of spacing properly the pressure taps on either side of the orifice plane by showing the general tendency of the coefficient of discharge to vary widely with the change of the taps, especially at the increased ratios

of orifice to pipe cross-section. The results, however, were not considered as final, as the tests were intended merely to furnish a general outline of the subject for investigation, and extreme accuracy was not sought for in the computations. From this outline plans were laid out for increasing the scope of the work to include larger sizes of pipes and also to establish more accurate standards of comparison.

THE FINAL INSTALLATION

13 On the removal of the company to its new headquarters in 1920, a research laboratory was established on the second floor of

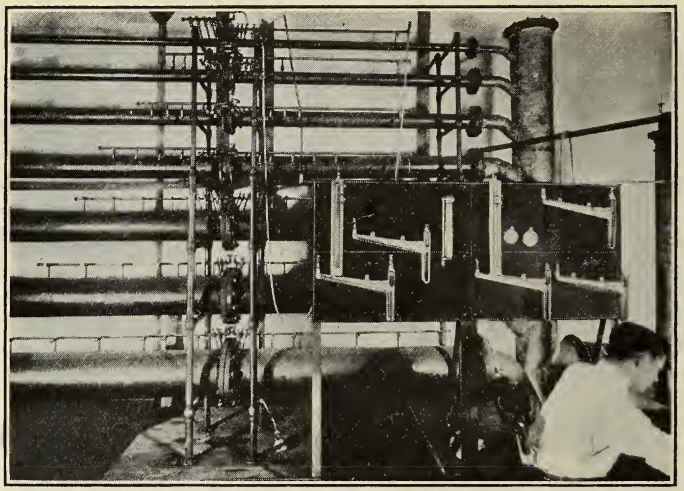


Fig. 4 Photographic View of Flanges, Pressure Taps and Gage Board

the factory building for the purpose of furthering the experimental work on orifice coefficients. Fig. 2 is a view of this laboratory and Fig. 3 shows the general arrangement of apparatus as originally installed.

14 The installation comprises several parallel piping units ranging in size from 2-in. to 12-in. standard and terminating at the ends in two 12-in. vertical manifolds. The second manifold opens into a horizontal outlet to which are attached five risers, each of which has an impact nozzle at the upper extremity.

15 The flow is produced by a No. 6 Sturtevant blower driven by a 15-hp. motor, and having a capacity of 6000 cu. ft. of air per

minute at a pressure of 10 in. of water. To vary the velocity of flow, an adjustable air gate was attached to regulate the intake to the blower.

16 Fig. 4 shows the extension of piping toward the second manifold and the impact nozzles. The flanges which support the orifice plates are shown with the pressure taps extending along the pipe on either side. The panel board in front of the piping supports the gages for measuring the various heads, a barometer, and a hygrometer. The vertical gages on the board were used only for heads of over 8 in. of water, which is above the range of the smaller extension type shown.

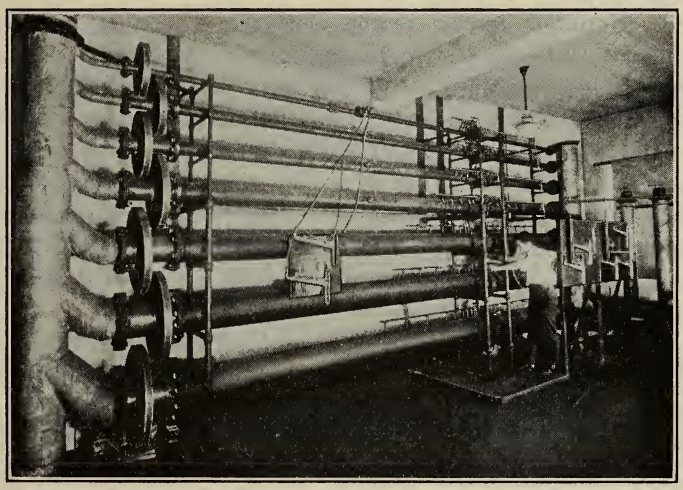


Fig. 5 Photographic View Showing Expansion Diaphragms

17 The pressure taps are controlled by special needle valves and terminate in the horizontal pressure manifolds that extend along the pipe. These manifolds also end in needle valves to which may be attached the rubber tubing to connect to the gages.

18 Fig. 5 is a view of the piping from the first manifold toward the flanges. The enlarged sections shown between the manifold and the piping are expansion diaphragms which permit a lateral movement of 1 in. without straining the joints of the piping. This arrangement is necessary in order to allow sufficient separation of the flanges for inserting the orifice plates.

19 The length of straight piping in front of the orifice plate,

exclusive of the diaphragms and inlets, is 12 ft. and beyond the plate 6 ft., not counting the inlets to the second manifold. These lengths were necessary to insure a straight-line flow ahead of the orifice and a complete restoration beyond it.

20 It was still possible, however, that the air as it entered the tubes would have a tendency to rush to the bottom of the pipe since the inlets opened downward; that is, in the direction of the flow in the manifold. To prevent this the inlets were partitioned inside by a series of vanes so arranged that part of the air was caught at the top and along the sides of the pipe. This gave the even distribution of the flow through the pipe section necessary for standard conditions.

21 Mounted on the piping is shown a smaller panel board carrying two Ellison 8-in. extension gages. One was used to determine the line pressure in the pipe at a point far enough in front of the orifice plate to be free from the effects of the interceptive flow; and the other to measure the differential of the pitot tube inserted in the pipe being tested so that the work could also include a study of pitot-tube coefficients. The tests were made with a bevel-edged pitot tube in which the openings faced in opposite directions and the edge of each made an angle of 70 deg. with the direction of the flow.

22 Fig. 2 shows the five risers at the further end of the room. The top of each of these risers is fitted with a pair of flanges and the opening in the upper flange is tapped to receive the nozzle.

23 The impact nozzle proper is made of monel metal. It was designed according to the proportions given by Mr. Wm. L. DeBaufre in his paper on the calibration of nozzles [1] presented at the Annual Meeting of the Society in 1920. Fig. 6 is a sample design showing the dimensions of tube No. 2.

24 Four sizes of openings are used and the nozzles are designated accordingly. No. 1 has an internal diameter of 1 in.; No. 2, of 2 in.; No. 3, of 3 in.; and Nos. 4 and 5 each with an internal diameter of 4 in.

25 To measure the velocity head of the air passing through the nozzle, $\frac{1}{4}$-in. bent tubes are used facing the flow and connected to the gages by rubber tubing, as in Fig. 6. These tubes are supported by means of a sliding seat fitted on a stem screwed into the side of the nozzle and can be adjusted vertically so that the dis-

[1] Calibration of Nozzles for the Measurement of Air Flowing into a Vacuum, Trans. Am.Soc.M.E., vol. 42, p. 641.

tance from the opening of the tube to the outlet of the nozzle may be varied from nothing to several inches.

26 The pressure at the base of the risers leading to the impact tubes is also recorded. This static pressure generally agreed with the velocity pressure of the impact, differing somewhat at higher velocities by the small friction drop through the risers.

27 Table 1 gives the distances in inches from the plane of the orifice to the pressure taps. There are twelve of these taps on each side, and thirteen on the downstream side for the smaller pipe sizes: seven placed closely together in the first diameter length, two in the second diameter length, and the remaining three at intervals of one

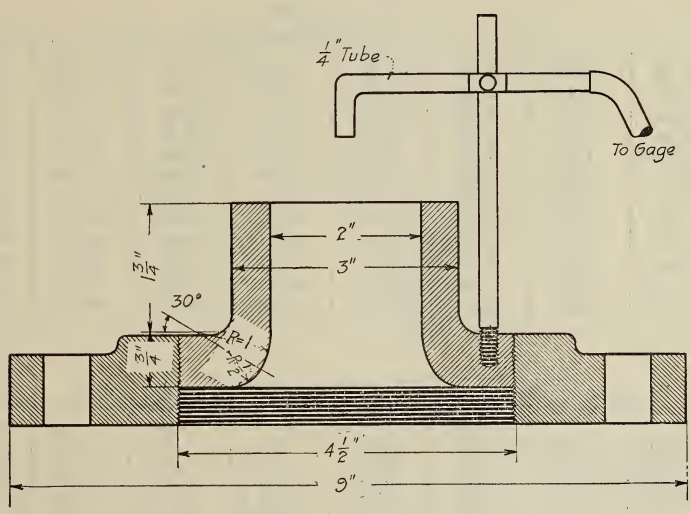


Fig. 6 Sample Design, Impact Nozzle No. 2

diameter length. These distances were chosen after preliminary tests had been made to set the points of possible variation of the differential pressure and also the limiting points of this variation.

28 Four thermometers are used to obtain the necessary temperature measurements. The first is placed in the first manifold and measures the temperature of the air as it comes to the pitot tube. The second is placed in the second manifold and gives the temperature of the flow in the line. The third is held over the opening of the impact nozzle, giving the temperature of the outcoming air. The fourth is on the board to measure the temperature of the ambient air.

29 The orifice plates were prepared from sheets of monel metal $\frac{1}{32}$ in. in thickness and were machined down to size to within a thousandth of an inch. The form shown in Fig. 7 was adopted to insure proper centering of the orifice. The two handles, placed

TABLE 1 DISTANCES IN INCHES FROM PLANE OF ORIFICE TO UPSTREAM AND DOWNSTREAM PRESSURE TAPS

No.	Approx. Diam. Length	Pipe Sizes in Inches						
		2	3	4	6	8	10	12
		Upstream (*A*) Taps						
1	0.05	0.28	0.28	0.44	0.52	0.66	0.66	0.88
2	0.10	0.38	0.34	0.59	0.61	0.86	1.00	1.17
3	0.20	0.41	0.66	0.72	1.11	1.46	2.08	2.45
4	0.30	0.59	0.94	1.28	1.83	2.44	3.00	3.70
5	0.50	1.00	1.47	2.00	3.05	4.08	5.03	6.09
6	0.75	1.53	2.28	2.97	4.58	6.08	7.48	9.11
7	1.00	2.03	3.00	3.93	6.07	8.14	10.06	12.11
8	1.50	3.03	4.56	5.96	9.14	12.16	15.05	18.17
9	2.00	4.10	6.15	7.96	12.14	16.22	20.12	24.09
10	3.00	6.20	9.15	12.02	18.24	24.35	30.25	36.33
11	4.00	8.20	12.15	16.02	24.34	32.48	40.38	48.38
12	5.00	10.20	15.25	20.08	30.44	40.58	50.45	60.50
		Downstream (*B*) Taps						
1	0.05	0.25	0.25	0.45	0.50	0.68	0.80	0.59
2	0.10	0.34	0.34	0.56	0.71	0.82	1.03	1.18
3	0.20	0.41	0.66	0.59	1.14	1.43	2.00	2.36
4	0.30	0.53	0.91	1.28	1.75	2.44	3.02	3.60
5	0.50	1.00	1.56	1.98	3.05	4.07	5.08	6.02
6	0.75	1.53	2.28	2.91	4.58	6.13	7.58	9.08
7	1.00	2.09	3.09	3.91	6.07	8.18	10.08	12.11
8	1.50	3.09	4.62	5.89	9.08	12.24	15.16	18.15
9	2.00	4.09	6.16	7.92	12.13	16.29	20.22	24.18
10	3.00	6.14	9.22	11.94	18.22	24.43	30.38	36.43
11	4.00	8.17	12.27	16.00	24.29	32.58	40.47	48.52
12	5.00	10.25	15.36	20.03	30.36	40.69	50.59	60.74
13	...	48.50	48.50	48.50	48.50	48.50	...	...

90 deg. apart, support guide pins so located that they will rest on two filed points on the flanges when the orifice is properly placed in the pipe. The figure also shows the method of centering and the measurements necessary.

METHOD OF TESTING

30 There are three classes of preparations to be considered before starting the test proper. They are as follows:

a The preparations for any one of the runs made on an orifice already installed, which are (1) the selection of standards, and (2) checking of the gages

b Preparations for a complete test of any given orifice, which include the preparation of the plate, besides the steps given under (*a*)

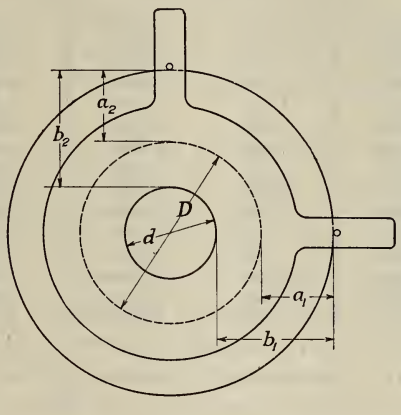


FIG. 7 METHOD OF CENTERING ORIFICES IN FLANGES

$a_1 + 0.5D = b_1 + 0.5d$ = distance to center of pipe; also $a_2 + 0.5D = b_2 + 0.5d$ = distance to center of pipe; hence $b_1 = a_1 + 0.5\ (D - d)$ and $b_2 = a_2 + 0.5\ (D - d)$. (d = orifice diameter; D = inside pipe diameter; a_1 and a_2 = radial distances from inside of pipe to point outside of flange; b_1 and b_2 = distances from orifice diameter to pins on handles.)

c The preparations for a set of tests on any given pipe which include (1) the examination of the pipe and flanges, (2) testing of the valves and connections, and also the steps in (*a*) and (*b*).

31 The complete work is actually carried on in the following way:

32 First, the pipe is spread apart at the flange and the inside wall thoroughly examined to make certain that there are no burrs on any one of the pressure openings and that the ends of the brass tubes are flush with the wall.

33 Two points are selected on the outside of the flange at

90 deg. apart and filed down to a measured radius in order to act as guides to secure proper centering of the orifice.

34 Second, the orifice to be tested is then selected and the edge of the opening examined carefully for burrs or chips. Any

TABLE 2 SIZES OF PIPES AND ORIFICES

Orifice Diam. d	Orifice Ratio d^2/D^2	Orifice Diam. d	Orifice Ratio d^2/D^2	Orifice Diam. d	Orifice Ratio d^2/D^2
12-in. Pipe, D = 12.08 in.		10-in. Pipe, D = 10.12 in.		8-in. Pipe, D = 8.12 in.	
2.707	0.0505	2.264	0.0500	1.809	0.0496
3.825	0.1002	3.190	0.0993	2.572	0.1033
4.699	0.1516	3.934	0.1510	3.158	0.1500
5.430	0.2012	4.529	0.2002	3.632	0.2000
6.050	0.2504	5.062	0.2502	4.017	0.2447
6.620	0.3000	5.539	0.2996	4.456	0.3011
7.120	0.3496	5.961	0.3470	4.780	0.3465
7.635	0.3995	6.400	0.4000	5.140	0.4000
8.108	0.4507	6.770	0.4475	5.452	0.4508
8.530	0.4992	7.153	0.4996	5.752	0.5018
8.930	0.5465	7.500	0.5491	6.021	0.5498
9.392	0.6045	7.826	0.5980	6.279	0.5979
9.800	0.6580	8.183	0.6528	6.588	0.6583
10.093	0.6895	8.424	0.6930	6.781	0.6974
6-in. Pipe, D = 6.06 in.		4-in. Pipe, D = 4.04 in.		3-in. Pipe, D = 3.06 in.	
1.360	0.0504	0.908	0.0505	0.686	0.0500
1.918	0.1002	1.277	0.1000	0.968	0.1000
2.374	0.1534	1.568	0.1506	1.191	0.1516
2.724	0.2025	1.814	0.2016	1.372	0.2010
3.030	0.2500	2.012	0.2480	1.530	0.2500
3.336	0.3030	2.228	0.3041	1.666	0.3070
3.577	0.3484	2.389	0.3497	1.818	0.3530
3.838	0.4011	2.547	0.3975	1.939	0.4015
4.090	0.4533	2.704	0.4480	2.042	0.4453
4.280	0.4994	2.857	0.5000	2.155	0.4960
4.500	0.5520	2.992	0.5485	2.256	0.5435
4.680	0.5980	3.117	0.5952	2.368	0.5988
4.903	0.6490	3.257	0.6500	2.485	0.6595
5.070	0.7000	3.385	0.7015	2.562	0.7010

roughness or dirt found on the edge is carefully removed without changing the size of the opening in any way.

35 The average diameter is then accurately determined to 0.001 in. with a vernier scale and recorded on the log sheet. The dimensions of the orifices for the various pipe sizes are given in Table 2. The distance b from the edge of the orifice to the guide

pin must be carefully measured and checked by the formulas given in Fig. 7, a in the formula being the radial distance from the inside of the pipe to the filed marks on the flange.

36 The orifice is then inserted in the pipe between two gaskets so that the two pins fall on the filed marks and all connections are made airtight.

37 Third, the gages used are of the Ellison 8-in. extension type. They must be checked for zero reading and proper leveling before any test is made. The rubber tubing is shellacked at all connections to cut down possibility of error.

38 Fourth, all the impact nozzles and all the gages are then sealed off with the exception of the differential gages across the orifice. The blower is started and allowed to build up its maximum head in the pipes. The valves are then tested for leakage in the following manner:

39 There are two sets of valves, those on the upstream side of the orifice being known as A-valves, numbered from 1 to 12, according to the tap which they control; and those on the downstream side or B-valves, numbered in like manner from 1 to 13. Valves A_1 and B_1 are opened and the gage reading noted. If either valve leaks to the atmosphere there will be a slight flow in the valve and, consequently, a reduced pressure on the differential column. If a zero reading is shown there is no leakage, or at most it is negligible. Then A_1 is left open, B_1 closed, and B_2, B_3, B_4, etc., opened and tested in like manner. Then all the A-valves are tested in the same way. If no suction or pressure is shown on the gage for any of the connections, the valves are airtight. If any leak was shown it was remedied by either tightening or replacing the stuffing box.

40 Fifth, the proper standards must now be selected. There are five impact nozzles to choose from, which vary in size from No. 1, which is 1 in. in diameter, to Nos. 4 and 5, which are 4 in. in diameter. The proper one to select is the one having as nearly as possible the same area as the orifice being tested. Since the flows through the impact nozzle and orifice are identical, the velocities through these sections will be the same. The head on the impact nozzle will therefore be close to that across the orifice, being somewhat different, of course, due to the difference in the discharge coefficients.

41 If the orifice is very large, two or more nozzles may be necessary. In this case the sum of the nozzle areas should equal the orifice area approximately. Equating the squares of the diameters serves the same end and is simpler than finding the area.

42 When the proper nozzle has been selected, the impact tube leading to the gage is adjusted to the proper distance from the nozzle opening and centered accurately with respect to this opening. From previous tests the proper distance to place the tube was found to be one-quarter of the nozzle diameter above the opening. For tube No. 1. a distance of $\frac{1}{2}$ in. is used, since to approach the opening would, in this case, have the effect of reducing the area of the opening and, consequently, of changing the coefficient.

43 Sixth, the blower is started with the gate wide open to determine the maximum head that could be obtained for the given orifice plate. This head is divided roughly into three or four steps to select the velocities at which the runs will be made. The blower intake is then reduced until the head on the impact-tube gage is reduced to the lowest velocity selected. The blower is allowed to run for a few minutes at this velocity before starting to take readings.

44 Seventh, valves A_1 and B_1 are then opened and the differential noted on the gage. B_2, B_3, etc., are opened successively until the highest differential is reached, indicating the position of the vena contracta in the flow. The readings at this point are recorded on the log sheet. Similar readings are taken for all the upstream taps, completing an A-run of the test. The impact pressure is recorded only once during each run, since it is kept constant by adjusting the intake gate to the blower. Temperature readings at the points previously mentioned are taken several times during each run. The barometer and hygrometer readings are taken and recorded for each run.

45 Eighth, three B-runs are then made at the same velocity, with the upstream taps at 7, 8 and 9, which were found always to give the same lowest static pressure on the upstream side, indicating the section of uniform flow in the pipe.

46 Ninth, another A-run is then made at the same velocity, with the downstream tap at B_{12}, which was found to give the highest static pressure on the downstream side, indicating the section of complete restoration of pressure in the line.

47 The other velocity steps are repeated in the same manner, thus completing a test on a given-ratio orifice.

CALIBRATION OF IMPACT NOZZLES

48 At the outset of the tests an attempt was made to calibrate the impact tubes, one at a time, by a number of diaphragm displacement meters. Three large meters, marked A, B, and C,

having a total capacity of 1000 cu. ft. per min., were connected together in such a manner that they could be run either in parallel or in series with each other and the system of piping leading to the impact nozzles.

49 A large number of tests (recorded in log sheets 1 to 199) have given fairly uniform results at the lowest capacity of the diaphragm meters, that is, at lower capacities the registration of the diaphragm meters indicated the same coefficient for a 10 per cent orifice in the 12-in. pipe and for the given impact nozzles, whether one, two, or three of these meters were used. At higher capacities the results varied greatly as compared with the flow indications. According to those results the coefficients of the impact nozzles were shown to decrease as the velocity through the tube increased, while from consideration of the effect of friction on the flow in the nozzle, it would appear that the coefficient should rather increase with increased velocity.

50 To verify the accuracy of the diaphragm meters a series of tests was made, placing meters *A*, *B*, and *C* in series with each other (log sheets 124 to 142) and then placing meters *A* and *B* in parallel against *C* in series at various capacities (log sheets 152 and 153). The results obtained showed a greater variation in the comparative registration of the meters than was permissible for their adoption as the standard of measurement.

51 From the data of these tests, when running the diaphragm meters at low capacities the indications corresponded very closely to the 10 per cent orifice in the 12-in. pipe, assuming the orifice coefficient to be 0.61. It was decided, therefore, to recalibrate the impact tubes against the 10 per cent orifice in the 12-in. pipe.

52 In calibrating each tube the method of procedure was first to determine the setting of the tubes on the impact nozzle at the flat part of the pressure curve, that is, at such a distance from the face of the nozzle that a small variation in the distance would not vary the reading of the gage for a given velocity of the air through the tube. It was found that the flat part was reached at one-quarter diameter from the face, with the exception of the 1-in. nozzle, where a $\frac{1}{2}$-in. minimum was required for this purpose. When the tube was placed at the proper setting, numerous runs were made at various velocities, the data were recorded on the log sheets properly indexed and filed, and the computations were made according to the standard method adopted for the test; that is, the quantity of air in pounds per minute was computed accurately for the orifice and for the im-

pact tubes, and the ratio of the two was recorded as the coefficient of the impact tube. Table 3 shows a sample test on impact tube No. 5 against the 10 per cent orifice in the 12-in. pipe, and Table 4 shows a similar test on the same tube with two orifices used for measuring the air.

TABLE 3 IMPACT-TUBE TEST

Log Sheet No. 1136. Tube No. 5: diameter, 4.004 in., $d^2 = 16.032$; setting, 1 in. Barometer, 29.62 in.; inches of water (p_o), 403; humidity per cent, 75; speed, 1800 r.p.m. Orifice standard, 10 per cent in 12-in. pipe; $K_n = 53.548$ for taps $A6$, $A7$, $B6$ and $B7$. Date, 5–24–22. Observers, A.F.S. & J.T.

No.	Pressures			Temperatures			Coefficients		
	Static h_s	12-in. Pipe h_n	Impact h_o	Impact t_1	Manifold t_m	Line t_b	G_o	G_n	C_v
1	0.14	0.590	0.200	75	76	79	11.13	11.63	0.957
2	0.21	0.895	0.305	..	..	..	13.72	14.36	0.956
3	0.27	1.160	0.395	..	..	..	15.61	16.35	0.955
4	0.34	1.495	0.505	76	77	80	17.74	18.49	0.960
5	0.41	1.760	0.595	..	..	..	19.24	20.10	0.957
6	0.48	2.080	0.700	..	..	..	20.90	21.75	0.961
7	0.55	2.380	0.800	76	77	80	22.37	28.23	0.961
8	0.63	2.690	0.905	77	78	81	23.80	24.73	0.962
9	0.70	2.970	1.000	..	..	..	24.97	26.00	0.961
10	0.77	3.290	1.100	..	..	..	26.30	27.30	0.963
11	0.84	3.600	1.200	77	78	81	27.50	28.50	0.965
12	0.91	3.890	1.300	78	79	81	28.60	29.70	0.963
13	0.98	4.220	1.400	..	..	..	29.80	30.80	0.967
14	1.05	4.490	1.500	..	..	..	30.70	31.85	0.964
15	1.12	4.830	1.600	78	79	81	31.85	32.90	0.968
16	1.19	5.090	1.700	..	..	..	32.70	33.77	0.968
17	1.26	5.380	1.795	..	..	..	33.60	34.70	0.968
18	1.33	5.700	1.900	78	79	81	34.60	35.70	0.969
19	1.40	6.010	2.000	..	..	..	35.60	36.60	0.973
20	1.47	6.310	2.100	79	80	82	36.40	37.54	0.969
21	1.54	6.620	2.200	..	..	..	37.30	38.40	0.972
22	1.62	6.910	2.300	..	..	..	38.10	39.30	0.970
23	1.69	7.200	2.400	79	80	82	38.90	40.10	0.970
24	1.76	7.520	2.500	..	..	..	39.80	40.90	0.973

IMPACT-TUBE COEFFICIENTS

53 The impact-tube coefficients as obtained in the final calibration are given in Figs. 8, 10, 11, 12, and 13, for tubes Nos. 1, 2, 3, 4 and 5. The points of calibration are reproduced on each curve to show the limits of accuracy in the observations. It will be noticed that while there is a considerable difference — as high as 1 per cent — in the actual value of the coefficient for the various sizes of tubes, all

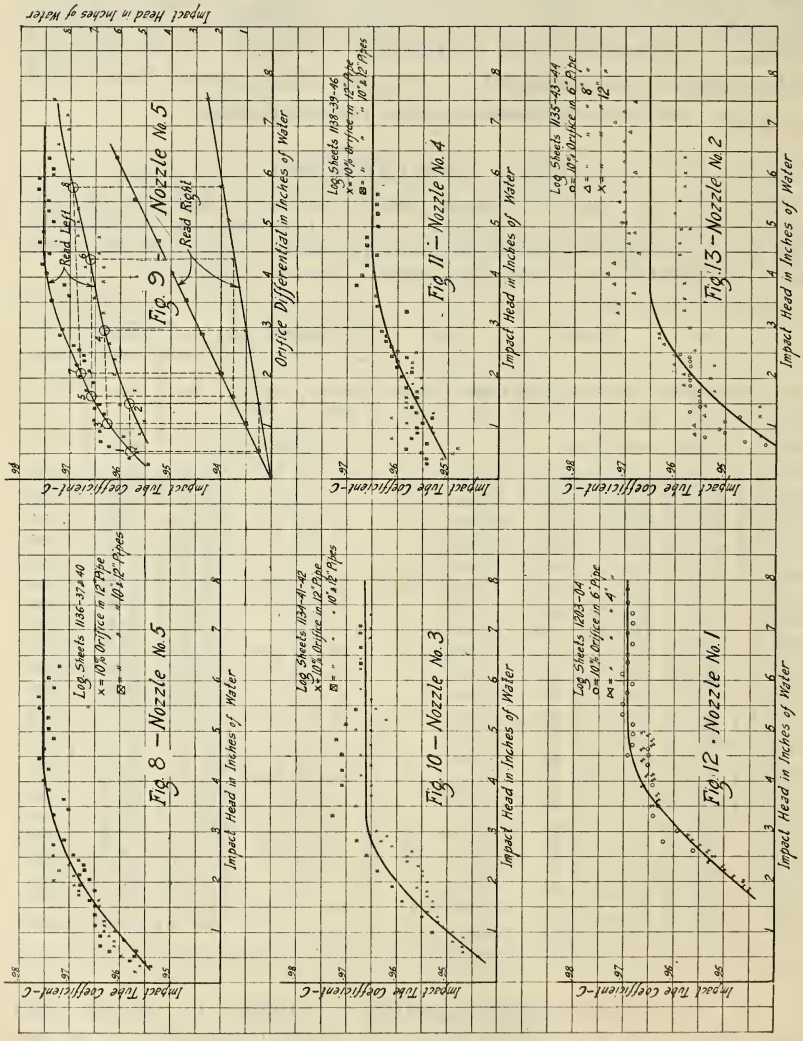
five curves show the same characteristic of sloping downward at the reduced impact pressure. The same shape of curve was obtained when plotting the results on the basis of orifice differential when the tests were limited to one tube and one orifice. The question naturally arose whether this characteristic was due to the tendency of the impact-tube coefficient to increase with increased head, or of the orifice coefficient to decrease with increased head.

TABLE 4 IMPACT-TUBE TEST

Log Sheet No. 1137. Tube No. 5: Diameter, 4.004 in.; $d^2 = 16\ 032$; setting, 1 in. Barometer, 29.53 in.; inches of water (p_o), 401.6; humidity per cent, 75; speed, 1800 r.p.m. Orifice standard 10 per cent in 12-in. pipe: $K_n = 53.548$ for taps $A7$, $B6$; orifice standard 10 per cent in 10-in. pipe; $K_n = 37.183$ for taps $A7$, $B6$. Date, 5–24–22. Observers, A.F.S. & J.T.

No.	Pressures				Temperatures			Coefficients		
	Static h_s	12-in. Pipe h_n	10-in. Pipe h_n	Impact h_o	Impact t_1	Manifold t_m	Line t_b	G_o	G_n	C_o
1	0.28	0.305	0.310	0.300	75	76	79	13.56	14.22	0.954
2	0.45	0.505	0.515	0.495	....	....	....	17.47	18.26	0.957
3	0.66	0.725	0.740	0.700	75.5	76.5	80	20.93	21.70	0.964
4	0.84	0.930	0.950	0.900	....	....	....	23.71	24.60	0.964
5	1.03	1.140	1.165	1.100	....	....	....	26.24	27.20	0.965
6	1.25	1.345	1.385	1.300	76.5	77.5	80.5	28.57	29.60	0.965
7	1.41	1.550	1.590	1.500	....	....	....	30.69	31.80	0.965
8	1.59	1.755	1.790	1.700	....	....	....	32.45	33.80	0.961
9	1.79	1.975	2.010	1.900	77	78	80.5	34.65	35.80	0.968
10	1.98	2.195	2.225	2.100	....	....	....	36.38	37.60	0.968
11	2.18	2.405	2.440	2.300	....	....	....	38.08	39.40	0.967
12	2.37	2.615	2.650	2.500	77	78	80.5	39.71	41.00	0.969
13	2.57	2.860	2.875	2.700	....	....	....	41.50	42.70	0.972
14	3.16	3.49	3.52	3.30	....	....	....	45.83	47.10	0.973
15	3.44	3.82	3.83	3.60	77	78	80.5	47.88	49.30	0.971
16	3.83	4.23	4.26	4.00	....	....	....	50.60	51.90	0.975
17	4.19	4.65	4.68	4.40	....	....	....	53.10	54.40	0.976
18	4.28	4.63	4.68	4.40	76	77	80	53.00	54.30	0.976
19	4.60	5.04	5.09	4.80	....	....	....	55.20	56.80	0.973
20	4.93	5.45	5.50	5.20	....	....	....	57.50	59.10	0.974
21	5.32	5.88	5.92	5.60	77	78	81	59.80	61.30	0.976
22	5.70	6.29	6.34	6.00	....	....	....	61.70	63.40	0.974

54 To determine this question, a number of similar tests were made with two orifice plates in parallel against one impact tube (see Table 4), thereby changing the relation between the impact head and the orifice differential for the same amount of flow. Fig. 9 shows the results of the tests on the No. 5 tube plotted on the basis of orifice differential, which are a reproduction of the same tests shown in Fig. 8 on the basis of impact head. It will be



FIGS. 8 TO 13 IMPACT-TUBE COEFFICIENTS

observed that while in Fig. 8 all the points of the tests are covered by one curve, those in Fig. 9 require two curves, following the difference in the impact head.

55 To demonstrate the case two other curves were drawn on the same chart, showing the impact head corresponding to each test. By tracing the points marked on the curves it will be noticed that the same coefficient is obtained for the same impact head regardless of the difference in the orifice head, thus demonstrating the fact that the variation in the coefficient is due to the effect of the change in velocity through the impact nozzle. This can be explained by the fact that the coefficient of friction increases considerably at very low velocities, which accounts for the reduced amount of flow through the impact nozzle.

56 The question further arose regarding the difference in the coefficients of the various tubes. At first an attempt was made to explain the difference by the possible error in the measurement of the diameter of the impact tube. This, however, could not be the case, as the vernier can be read easily within a thousandth of an inch, which, in the case of the Nos. 4 and 5 tubes, would make the error entirely insignificant. Numerous tests were repeated for the purpose of ascertaining these discrepancies, and in every case there was shown to be a difference of about .1 per cent between the coefficients on the Nos. 4 and 5 tubes, both having the same internal diameter of approximately 4 in.

57 From the viewpoint of approach the No. 4 tube is farther away from the bend of the manifold than the No. 5 tube. When the two of them were open together, No. 4 would show a higher differential than No. 5, from which it appears that the shape of the approach to the tube has some effect upon the coefficient of discharge. To eliminate this effect it would appear that the approach to the impact tube should be made of larger relative size and at a longer distance from an elbow or bend in the pipe.

THEORETICAL RELATION OF FUNCTIONS

58 The following notation is adopted with reference to Fig. 1, in which most of the quantities and dimensions are demonstrated diagrammatically.

d = diameter of the thin-plate orifice, either in the restricted pipe or in the wall of the large vessel

D = diameter of pipe

X = percentage of jet contraction at the smallest cross-section or the vena contracta of stream. (Shown as x in Figs. 1, 14 and 15.)

r = ratio of orifice cross-section to pipe cross-section, or d^2/D^2; Xr = ratio of vena contracta to cross-section of pipe

ρ = $1/r$ = ratio of pipe cross-section to orifice cross-section or D^2/d^2; ρ/X = ratio of pipe cross-section to the vena contracta of stream

A = distance from plane of orifice to upstream pressure taps

B = distance from plane of orifice to downstream pressure taps

h_1 = static pressure at section of normal flow ahead of orifice

h_2 = static pressure at section of vena contracta

h_3 = static pressure at section of normal flow beyond orifice

h = differential head between any two points on opposite sides of the orifice

h_v = differential head between the two points designated as h_1 and h_2. This quantity represents the pressure drop across the orifice corresponding to the increased velocity of the flow

h_p = differential head between the two points designated as h_1 and h_3. This quantity measures the total loss of pressure caused by the sudden contraction and expansion of the flow through the orifice

Z = ratio of h_p to h_v

V = velocity of fluid at section of normal flow in pipe

v = velocity of fluid at vena contracta of stream

Xv = velocity of fluid at section of orifice

Q = rate of flow or amount of fluid passing through the orifice in a given unit of time

C = coefficient of discharge defined by —

$$Q = C\frac{\pi}{4}\frac{d^2}{\sqrt{1-r^2}}\sqrt{2gh} = C\frac{\pi}{4}\frac{D^2}{\sqrt{\rho^2-1}}\sqrt{2gh}$$

C' = velocity coefficient of orifice defined by —

$$v = C'\sqrt{2gh}, \text{ and } C' = f(A)f(B)\sqrt{\frac{1}{1-X^2r^2}}$$

(XC') = orifice coefficient defined by —

$$Q = (XC')\frac{\pi}{4}d^2\sqrt{2gh}$$

C'' = orifice coefficient defined by —

$$Q = C''\frac{\pi}{4}D^2\sqrt{2gh}, \text{ or } C'' = (XC')r$$

59 Subscript v, with reference to the given terms, applies to the condition of pressure tap A at normal flow ahead of the orifice plane and pressure tap B at the vena contracta. Subscript p applies to the condition of pressure tap A at normal flow ahead of the orifice plane, and pressure tap B at normal flow after the orifice plane.

60 In the case of the frictionless orifice in the large vessel the amount of fluid Q passing through the opening in a unit of

time is a function of the three variables: h, the differential head; d, the diameter of the opening; and X, the contraction of the jet upon leaving the vessel, or —

$$Q = f(X, d, h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

61 In the case of the pipe orifice, the rate of flow Q is a function of six variables: namely, the differential head h, the diameter of the orifice d, the jet contraction X, the orifice ratio r, and the high- and low-pressure distances A and B, respectively, or —

$$Q = f\,(X, d, h, r, A, B) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

For the first case we may write the fundamental equation —

$$\frac{v^2}{2g} = h, \text{ or } v = \sqrt{2gh}$$

and

$$Q = \frac{\pi}{4} X d^2 v = \frac{\pi}{4} X d^2 \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

62 Equation [3] offers the complete development of Equation [1]. In this case X is the ratio of the actual amount of fluid passing through the jet and the theoretical amount corresponding to the differential head and the given size of the opening. If we assume a given test for determining this ratio, we obtain from the test the actual value of Q and the known values of d and h, so that—

$$X = \frac{Q}{\frac{\pi}{4} d^2 \sqrt{2gh}} = \text{coefficient of discharge} \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

63 For the case of the pipe orifice, the fundamental equation will refer primarily to the velocities V and v, and the corresponding heads h_1 and h_2, taking the form of —

$$\frac{V^2}{2g} + h_1 = \frac{v^2}{2g} + h_2$$

or —

$$\frac{v^2 - V^2}{2g} = h_1 - h_2 = h_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

and since —

$$D^2 V = X d^2 v$$

therefore —

$$V = X \frac{d^2}{D^2} v = Xrv$$

Substituting in [5],

$$(1 - X^2 r^2) \frac{v^2}{2g} = h_v$$

or —

$$v = \sqrt{\frac{1}{1 - X^2r^2}}\ \sqrt{2gh_v}$$

and —

$$Q = \frac{\pi}{4} Xd^2v = \frac{\pi}{4} Xd^2\sqrt{\frac{1}{1 - X^2r^2}}\ \sqrt{2gh_v} \quad . \quad . \quad . \quad . \quad . \quad [6]$$

64 Equation [6] offers the development of the functional expression [2] only with respect to the variables X, d, h, and r at the particular sections at which h_v is obtained. To make the equation general, we introduce $f(A)$ and $f(B)$ to take care of the effect of varying the location of the taps, so that, for any value of h,

$$Q = f(A)\, f(B)\, \frac{\pi}{4} Xd^2\sqrt{\frac{1}{1 - X^2r^2}}\ \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where $f(A)$ and $f(B)$ are equal to unity when h_v is substituted for h.

65 If we now assume a given test for the purpose of determining the experimental functions of Equation [7], we obtain, on one hand, the value of Q, and on the other hand, the given values of d, h, and r, so that —

$$f(A)\, f(B)\, X\sqrt{\frac{1}{1 - X^2r^2}} = \frac{Q}{\frac{\pi}{4} d^2\sqrt{2gh}} \quad . \quad . \quad . \quad . \quad . \quad [8]$$

or —

$$(XC') = \frac{Q}{\frac{\pi}{4} d^2\sqrt{2gh}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

where —

$$(XC') = X f(A)\, f(B)\sqrt{\frac{1}{1 - X^2r^2}} \quad . \quad . \quad . \quad . \quad . \quad [9a]$$

and —

$$Q = (XC')\frac{\pi}{4} d^2\sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

66 That is, in the case of the pipe orifice, the experimental coefficient XC' is a function of the jet contraction X, the orifice ratio r, and the high- and low-pressure distances A and B, respectively. This function necessarily approaches unity as a limit when the conditions of the pipe orifice approach the condition of the rounded opening in a large vessel.

67 Equation [7] is often replaced by the form generally used for the venturi tube:

$$Q = C\frac{\pi}{4}\frac{d^2}{\sqrt{1 - r^2}}\sqrt{2gh} = C\frac{\pi}{d}\frac{D^2}{\sqrt{\rho^2 - 1}}\sqrt{2gh} \quad . \quad . \quad [11]$$

68 The quantity C in Equation [11] is the coefficient of discharge for the orifice as compared with the coefficient of discharge for the venturi tube. In using this coefficient, it is always necessary when solving Equation [11] to compute the radical $\sqrt{1 - r^2}$ for the given orifice ratio. This is especially troublesome when a solution is sought for the size of the orifice necessary for a given differential and amount of flow, as it involves two unknown quantities C and r. To overcome this difficulty, a modification of Equation [10] is employed with reference to the cross-section of the pipe. Since $d^2 = D^2r$, we can write —

$$Q = (XC')r \frac{\pi}{4} D^2 \sqrt{2gh}$$

or —

$$Q = C'' \frac{\pi}{4} D^2 \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11a]$$

Equation [11a] in combination with a $C'' - r$ curve gives a direct solution for the size of orifice necessary for a given flow.

69 The pressures h_1, h_2, and h_3 at the three main sections of the orifice region are significant for the fact that they indicate the relation between the drop of pressure h_v equivalent to the increase in velocity and the pressure loss h_p, caused by the sudden contraction and expansion of the flow. From the above equations, we can write —

$$C_p \sqrt{h_p} = C_v \sqrt{h_v}$$

from which —

$$\text{Fraction of loss } Z = \frac{h_p}{h_v} = \left(\frac{C_v}{C_p}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad [12]$$

and fraction of pressure restoration $= 1 - Z = 1 - \frac{h_p}{h_v} = 1 - \left(\frac{C_v}{C_p}\right)^2$.

The same applies to any of the experimental coefficients determined for the two sections corresponding to h_2 and h_3, respectively.

70 In calculating the data of a given test it is more convenient to obtain the quantity XC' directly as given in Equation [9] without any reference to the orifice ratio. Having obtained this quantity, the other functions can be easily computed from the known relations to XC'.

71 From the above equations we derive the values of X, C, and C'' in term of XC' as follows:

$$(XC') = X f(A) f(B) \sqrt{\frac{1}{1 - X^2r^2}} \quad . \quad . \quad . \quad . \quad [9a]$$

Solving for X,

$$X = \frac{(XC')}{\sqrt{\{f(A)\,f(B)\}^2 + (XC')^2\,r^2}} \quad . \; . \; . \quad [13]$$

At the vena contracta $f(A)\,f(B) = 1$. Therefore —

$$X = \frac{(XC')_v}{\sqrt{1 + (XC')_v^2\,r^2}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [13a]$$

Combining Equations [10] and [11],

$$(XC') = \frac{C}{\sqrt{1 - r^2}}$$

or —

$$C = (XC')\sqrt{1 - r^2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [14]$$

From Equations [10] and [11a],

$$(XC')\,d^2 = C''D^2$$

from which —

$$C'' = (XC')\frac{d^2}{D^2} = (XC')r \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [15]$$

COMPUTATION OF RESULTS

72 The method adopted was to compute the rate of flow in pounds per minute for the calibrated impact nozzles used as a standard, and the rate in pounds per minute derived theoretically for the given orifice plate, assuming XC' to be equal to unity. The actual value of the coefficient XC' for the given orifice is necessarily the ratio of the former to the latter. The method of procedure is demonstrated on the sample log sheets (Tables 5 and 6) which contain the data of one A-run and one B-run on the 12-in. pipe. The symbols designating the data and results on the log sheets refer to the following notation:

Pressure in Inches of Water:

h_m = last manifold pressure (entrance to impact nozzles)
h_b = line pressure, first manifold
h_d = pitot-tube kinetic pressure
h_n = orifice differential
h_e = impact-tube differential
h_a = pitot-tube differential
h_s = static pressure, downstream orifice tap

Temperature in Degrees Fahrenheit:

t_o = impact-jet temperature
t_r = room temperature
t_m = last manifold temperature
t_b = line temperature, first manifold

Density of Fluid, Pounds per Cubic Foot:

γ_o = density of air at outlet of impact nozzle

γ_n = density of air at downstream side of orifice plate

γ_b = density of air at location of pitot tube

Diameter in Inches:

d_o = impact-tube diameter

d_n = orifice diameter

D = pipe diameter

TABLE 5 ORIFICE TEST

Log Sheet No. 574. Pipe diam. = 12.08 in.; D^2 = 145.926; ratio 50.21 per cent; Barometer, 29.28 in.; inches of water (p_o), 398; humidity per cent, 45; speed, 1800 r.p.m.; Tube No. 5: $d_o = 4.004$ in.; $d_o^2 = 16.032$; $K_o = 93.60$; $K_n = 43.89$; $K_b = 874.099$. Tube No. 4: $d_o^2 = 3.998$ in., $d_o^2 = 15.984$; $K_o = 92.40$. Date, 4–12–22.

Conn. B_5	Sta. h_s	Dif. h_n	K_a per h_n	$\sqrt{y_n}$, h_s & t_m	$\sqrt{h_n}$	G_n	$C_n = G/G_n$
A1	4.08	1.56	0.998	2716	1.249	148.6	0.682
2	4.08	1.55	0.998	2716	1.245	148.2	0.684
3	4.08	1.53	0.998	2716	1.237	147.3	0.688
4	4.08	1.51	0.998	2716	1.229	146.3	0.693
5	4.08	1.49	0.999	2716	1.221	145.4	0.698
6	4.08	1.48	0.999	2716	1.217	144.8	0.702
7	4.08	1.48	0.999	2716	1.217	144.8	0.702
8	4.08	1.48	0.999	2716	1.217	144.8	0.702
9	4.08	1.48	0.999	2716	1.217	144.8	0.702
10	4.08	1.48	0.999	2716	1.217	144.8	0.702
11	4.08	1.48	0.999	2716	1.217	144.8	0.702
12	4.08	1.49	0.999	2716	1.221	145.4	0.698

Impact Tube	No. 5	No. 4	Pitot Tube	
1. Impact (h_o)	4.00	4.00	1. Line Pressure (h_b)	5.55
2. K_a (per h_o)	0.998	0.998	2. Dynamic (h_d)	5.82
3. $\sqrt{y_o}$ (p_o & t_o)	0.2705	0.2705	3. Tube Diff. (h_a) 0.34 $\sqrt{h_a}$	0.583
4. $\sqrt{h_o}$	2.000	2.049	4. K_a (per h_d–h_b)	1.000
5. $G_o = K_o\ K_a\ \sqrt{y_o}\ \sqrt{h_o}$	50.44		5. $\sqrt{y_b}$ (p_o–h_b & t_b)	0.2718
6. $G_o = K_o\ K_a\ \sqrt{y_o}\ \sqrt{h_o}$	51.04		6. $G_b = K_b\ K_a\ \sqrt{y_b}\ \sqrt{h_a}$	138.5
7. G	101.48		7. $C_b = G/G_b$	732

$Q = G/0.0764 = 1328$; $= t_o = 71$; $t_m = 72$; $t_b = 73$; $h_m = 4.12$.

Factors (constant for a given test):

$K_o = 5.99\ C_o d_o^2$ = impact-tube factor

$K_b = 5.99\ D^2$ = pitot-tube factor

K_a = thermodynamic-expansion factor

$K_n = 5.99\ d_n^2$ = orifice factor

Rate of Flow in Pounds of Air per Minute:

$G_o = K_o K_a \sqrt{h_o \gamma_o}$ for impact tube

$G_b = K_b K_a \sqrt{h_b \gamma_b}$ for pitot tube

$G_n = K_n K_a \sqrt{h_n \gamma_n}$ for orifice plate

Experimental Coefficients:

C_o = impact-tube coefficient

C_n = orifice coefficient $XC' = \frac{G_o}{G_n}$

C_b = pitot-tube coefficient $= \frac{G_o}{G_b}$

TABLE 6 ORIFICE TEST

Log Sheet No. 575. Pipe diam., 12.08 in.; D^2 = 145.926; ratio, 50.21 per cent; Barometer, 29.28 in.; inches of water (p_o), 398; humidity per cent, 45; speed, 1800 r.p.m. Tube No. 5: d_o = 4.004 in.; d_o^2 = 16.032; K_o = 93.60; K_n = 43.891; K_b = 874.099. Tube No. 4: d_o = 3.998 in.; d_o^2 = 15.984; K_o = 92.40. Date, 4-12-22.

Conn. A_7	Sta. h_s	Dif. h_n	K_a per h_n	$\sqrt{y_n}$, h_s & t_m	$\sqrt{h_n}$	G_n	$C_n = G/G_n$
$B1$	4.11	1.45	0.999	0.2713	1.204	143.2	0.708
2	4.10	1.46	0.999	0.2713	1.208	143.6	0.706
3	4.10	1.46	0.999	0.2713	1.208	143.6	0.706
4	4.09	1.47	0.999	0.2713	1.212	144.1	0.703
5	4.08	1.48	0.999	0.2713	1.217	144.6	0.702
6	4.12	1.44	0.999	0.2713	1.200	142.7	0.711
7	4.24	1.32	0.999	0.2713	1.149	136.6	0.741
8	4.48	1.08	0.999	0.2714	1.039	123.5	0.820
9	4.64	0.92	0.999	0.2714	0.959	114.0	0.888
10	4.77	0.79	0.999	0.2715	0.889	105.9	0.961
11	4.81	0.75	0.999	0.2715	0.866	103.0	0.983
12	4.82	0.74	0.999	0.2715	0.860	102.2	0.991

Impact Tube	No. 5	No. 4
1. Impact (h_o)	4.00	4.20
2. K_a (per h_o)	9.98	9.98
3. $\sqrt{y_o}$ (p_o & t_o)	2.702	2.702
4. $\sqrt{h_o}$	2.000	2.049
5. $G_o = K_o\ K_a\ \sqrt{y_o}\ \sqrt{h_o}$	50.4	
6. $G_o = K_o\ K_a\ \sqrt{y_o}\ \sqrt{h_o}$	51.0	
7. G	101.4	

Pitot Tube	
1. Line Pres. (h_b)	5.55
2. Dynamic (h_d)	587
3. Tube Diff. (h_a) 0.34 [$\sqrt{h_a}$ = 0.583]	
4. K_a (per h_d–h_b)	1.000
5. $\sqrt{y_b}$ (p_o–h_b & t_b)	0.2781
6. $G_b = K_a\ \sqrt{y_b}\ \sqrt{h_a}$	138.5
7. $C_b = G/G_b$	0.732

$Q = G/0.0764 = 132.8$; $t_o = 72$; $t_m = 73$; $t_b = 74$; $h_m = 4.12$.

73 The quantity $K = 5.99d^2$ is arrived at by modifying Equation [10] to signify h in inches of water, d in inches, and replacing $Q\gamma$ by G. The quantity K_a is given in Table 7 for the various pressure and orifice ratios. The figures of the table are arrived at by comparing the thermodynamic solution of the flow with the hydraulic solution for the same pressure difference.

74 A large number of additional tables were prepared to facilitate the computations in general. One was prepared for each size of pipe giving the constants d^2, K_n, r, r^2, etc., for the fourteen orifices of the given pipe. Another table was made out

for the values of γ and $\sqrt{\gamma}$ at varying pressures and temperatures to cover the range of the tests. A set of curves was prepared giving the values of K_o for each tube corresponding to the variations of the impact head. With the aid of these tables and curves, the results of the tests were computed immediately at the end of each run and were recorded on the same log sheet.

75 The data of the final tests are recorded on 1625 log sheets filed in chronological order and cross-indexed under the various headings of impact tube, pipe size, and orifice ratio. A log sheet of the impact-tube tests contains a run on a given tube either varying the velocity head at constant setting or the setting at constant

TABLE 7 VALUES OF THE FACTOR K_a CORRESPONDING TO PRESSURE DIFFERENCE IN INCHES OF WATER

Inches of Water	Orifice Ratios											
	10	20	25	30	35	40	45	50	55	60	65	70
1	1.000	0.999	0.999	0.999	0.999	0.999	0.999	0.999	0.999	0.998	0.998	0.998
2	0.999	0.999	0.999	0.998	0.998	0.998	0.998	0.998	0.998	0.997	0.997	0.996
3	0.999	0.998	0.998	0.998	0.997	0.997	0.997	0.997	0.996	0.996	0.995	0.993
4	0.998	0.998	0.998	0.997	0.997	0.997	0.996	0.996	0.995	0.994	0.993	0.991
5	0.998	0.997	0.997	0.997	0.996	0.996	0.995	0.995	0.994	0.993	0.991	0.989
6	0.997	0.997	0.997	0.996	0.996	0.995	0.994	0.994	0.993	0.992	0.989	0.987
7	0.997	0.996	0.996	0.995	0.995	0.995	0.994	0.993	0.992	0.990	0.987	0.985
8	0.996	0.996	0.996	0.995	0.995	0.994	0.993	0.992	0.991	0.989	0.986	0.983
9	0.996	0.995	0.995	0.994	0.994	0.993	0.992	0.991	0.989	0.987	0.984	0.981
10	0.995	0.995	0.995	0.994	0.993	0.992	0.991	0.990	0.988	0.986	0.982	0.979
11	0.995	0.994	0.994	0.993	0.992	0.991	0.990	0.989	0.987	0.984	0.980	0.977
12	0.994	0.994	0.993	0.992	0.991	0.990	0.989	0.988	0.985	0.983	0.979	0.974

NOTE: One inch of water corresponds to $(P_2 - P_1)/P_1 = 1/400$.

velocity. A log sheet of the orifice tests contains a run on a given-ratio orifice at constant velocity either varying the upstream A-taps with the downstream constant, or the downstream B-taps with the upstream constant. A complete test for a given-orifice ratio comprises four velocity tests and from five to twelve log sheets for each velocity, part of them with A constant and B varying, and part with B constant and A varying. Each pipe size was tested for fourteen orifice ratios with the exception of the 2-in. pipe, on which the lower-ratio orifices were not completed at the writing of this paper.

76 The log sheets for the higher-ratio tests also included the readings of the pitot-tube differential for the given velocity. From

these readings the theoretical rate of flow was obtained and the coefficient of the pitot tube was computed for the given-pipe size. This coefficient was found to have a fairly constant value, 0.73, for the 6-in., 8-in., 10-in., and 12-in. pipes. In the smaller sizes the coefficient was reduced considerably, showing a value of 0.70 for the 4-in. pipe and 0.66 for the 3-in. pipe. It was also found that the maximum differential was obtained when the beveled opening of the tube was turned at an angle of about 10 deg. from the direction of the flow. This phenomenon could be explained by the increased siphoning action on the complementary opening facing in the opposite direction of the flow.

CHARACTERISTIC CURVES OF ORIFICE PLATES

77 At the completion of a test for a number of orifice plates in a given size of pipe the values of the coefficient XC' were plotted in three sections on large sheets ruled in logarithmic coördinates. The first section represented the so-called A-curves plotted as a function of the upstream distances from the A-runs of the test with the downstream connection at the vena contracta. The second section represented the B-curves plotted as a function of the downstream distances from the B-runs of the test. The third section were A-curves plotted as a function of the upstream distances with the downstream connection at four and one-half diameters of the pipe or the point of complete restoration of pressure.

78 These curves, not given in the paper, represented in substance the form similar to the three main sets of the final curve, shown on Fig. 15 for the 6-in. pipe, except that in the original curves each orifice ratio was represented by a group of lines adjoining the test points of the several velocities for a given ratio. Also in those curves the actual orifice ratios were used, which differed somewhat from the equal divisions of the final curves in Fig. 15. With small exceptions the lines of each group did not deviate much from each other, so that in each case a single line was drawn which represented the average value of the coefficient for the given ratio.

79 From these average lines the points of minimum and maximum were plotted on Fig. 14 as a function of the orifice ratio for the various sizes of pipe, the lower set for the coefficient $(XC')_v$ at the vena contracta, and the upper set for the coefficient $(XC')_p$ at four and one-half diameters from the orifice plate. After the points of the two sets on Fig. 14 were joined by smooth lines for

the given sizes of pipes, the corrected values were obtained for the even ratios in multiples of five and the points transferred to the

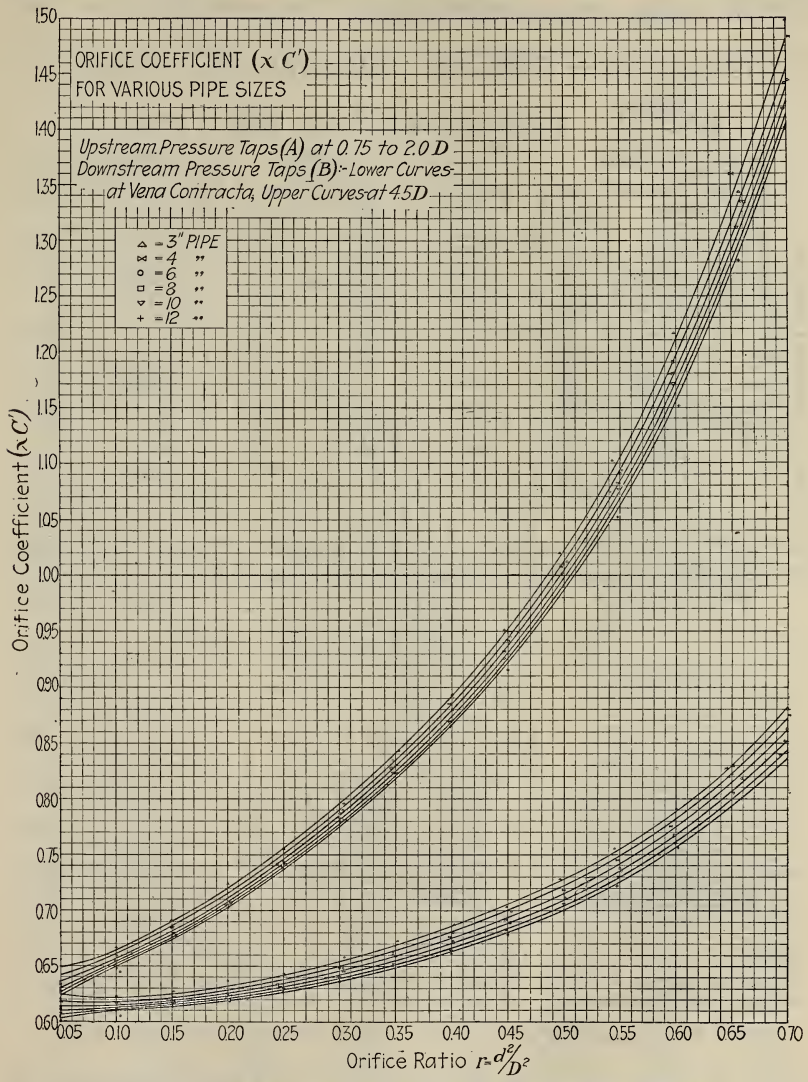


FIG. 14 ORIFICE COEFFICIENTS FOR VARIOUS PIPE SIZES

original curves and compared with the average lines of the nearest ratio. A new line was drawn parallel to the old one to pass through the corrected points for the even ratio. In this manner the effects

of possible errors of observation were reduced, first, by comparison with the similar tests of the same ratio, second, by comparison with the similar tests of various ratios of the same size of pipe, and

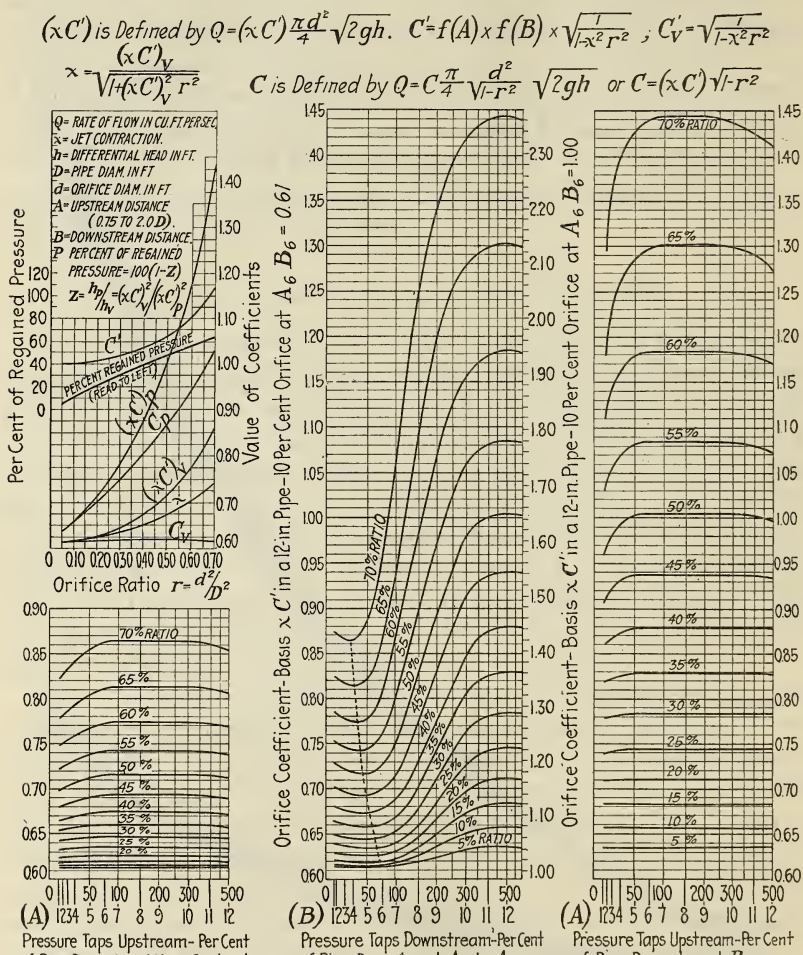


FIG. 15 CHARACTERISTIC CURVES FOR ORIFICE PLATES, 6-IN. PIPE

third, by comparison with the similar tests of the various sizes of pipes.

80 The final curves for the even ratios of the 6-in. pipe are shown in the three main sections of Fig. 15. These three sections give a generalization of the complete work for the given size of

pipe. They point out forcibly the great importance that must be attached to the proper spacing of the orifice connections. In fact,

TABLE 8 ORIFICE COEFFICIENTS FOR 12-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.603	0.624	0.603	1.000	0.603	0.624	0.067
10	0.610	0.650	0.609	1.002	0.607	0.647	0.119
15	0.614	0.676	0.612	1.003	0.607	0.668	0.174
20	0.620	0.703	0.615	1.008	0.608	0.689	0.223
25	0.627	0.736	0.619	1.013	0.608	0.714	0.274
30	0.637	0.774	0.626	1.018	0.608	0.739	0.326
35	0.649	0.818	0.633	1.025	0.608	0.767	0.371
40	0.663	0.867	0.641	1.034	0.608	0.795	0.416
45	0.679	0.923	0.649	1.046	0.606	0.824	0.459
50	0.700	0.987	0.661	1.059	0.606	0.855	04.97
55	0.725	1.063	0.674	1.076	0.605	0.887	0.535
60	0.756	1.157	0.689	1.097	0.605	0.926	0.573
65	0.795	1.273	0.706	1.126	0.605	0.968	0.610
70	0.846	1.413	0.728	1.162	0.604	1.010	0.641

they would seem to indicate that an excessive precision is necessary in determining these points, since a very small change in the tap

TABLE 9 ORIFICE COEFFICIENTS FOR 10-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.606	0.627	0.606	1.000	0.606	0.627	0.066
10	0.611	0.652	0.610	1.002	0.608	0.649	0.122
15	0.615	0.678	0.612	1.005	0.608	0.670	0.177
20	0.621	0.705	0.616	1.008	0.609	0.691	0.224
25	0.630	0.738	0.622	1.013	0.611	0.715	0.271
30	0.640	0.777	0.629	1.018	0.611	0.742	0.321
35	0.651	0.821	0.634	1.026	0.610	0.769	0.371
40	0.665	0.870	0.642	1.035	0.610	0.797	0.416
45	0.683	0.926	0.653	1.046	0.610	0.827	0.456
50	0.704	0.992	0.664	1.060	0.610	0.859	0.496
55	0.730	1.069	0.677	1.074	0.609	0.893	0.534
60	0.761	1.165	0.692	1.099	0.604	0.932	0.573
65	0.800	1.281	0.710	1.127	0.608	0.974	0.610
70	0.851	1.420	0.728	1.163	0.608	1.014	0.641

distance along the steeper parts of the curves might cause a change of several per cent in the value of the coefficient.

81 To eliminate possibility of large errors in the practical

application of the orifice, it is best, therefore, to select only these points for use at which a slight change in the position of the tap

TABLE 10 ORIFICE COEFFICIENTS FOR 8-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.610	0.631	0.610	1.000	0.610	0.631	0.065
10	0.613	0.654	0.611	1.003	0.610	0.651	0.120
15	0.618	0.680	0.615	1.005	0.611	0.672	0.175
20	0.624	0.708	0.619	1.008	0.612	0.694	0.223
25	0.632	0.741	0.624	1.013	0.613	0.718	0.272
30	0.643	0.780	0.631	1.019	0.614	0.744	0.320
35	0.655	0.825	0.638	1.027	0.614	0.773	0.369
40	0.670	0.874	0.646	1.035	0.614	0.801	0.412
45	0.688	0.931	0.657	1.047	0.614	0.831	0.454
50	0.710	0.997	0.669	1.061	0.614	0.863	0.493
55	0.736	1.076	0.682	1.079	0.614	0.898	0.532
60	0.767	1.173	0.697	1.100	0.614	0.939	0.572
65	0.806	1.290	0.714	1.129	0.613	0.982	0.609
70	0.856	1.430	0.734	1.166	0.612	1.021	0.641

would not appreciably affect the coefficient. This condition is obviously fulfilled at the minimum and maximum points, since these

TABLE 11 ORIFICE COEFFICIENTS FOR 6-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.615	0.636	0.615	1.000	0.615	0.636	0.065
10	0.615	0.657	0.614	1.002	0.612	0.654	0.123
15	0.620	0.683	0.618	1.003	0.613	0.675	0.175
20	0.626	0.711	0.621	1.008	0.614	0.698	0.225
25	0.635	0.745	0.627	1.013	0.615	0.722	0.273
30	0.646	0.784	0.634	1.020	0.616	0.748	0.320
35	0.659	0.829	0.642	1.027	0.618	0.777	0.368
40	0.674	0.880	0.650	1.037	0.618	0.807	0.413
45	0.693	0.938	0.662	1.047	0.619	0.838	0.455
50	0.716	1.004	0.674	1.062	0.620	0.870	0.487
55	0.742	1.084	0.687	1.081	0.619	0.905	0.531
60	0.774	1.183	0.702	1.103	0.619	0.947	0.572
65	0.813	1.301	0.719	1.131	0.618	0.990	0.609
70	0.864	1.444	0.739	1.170	0.617	1.031	0.642

points occur at the flat portion of the curve.

82 The A-curves given are restricted to a study of conditions at these two points. These two sets can be used in a similar manner

to select the proper points at which to place the upstream pressure taps. Thus, since the flat parts of the curve occur between points

TABLE 12 ORIFICE COEFFICIENTS FOR 4-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.619	0.642	0.619	1.000	0.619	0.642	0.070
10	0.617	0.661	0.616	1.002	0.614	0.658	0.130
15	0.622	0.686	0.620	1.003	0.615	0.678	0.177
20	0.628	0.716	0.624	1.007	0.616	0.702	0.231
25	0.638	0.750	0.630	1.013	0.618	0.727	0.276
30	0.650	0.790	0.638	1.019	0.620	0.754	0.323
35	0.662	0.835	0.645	1.027	0.621	0.782	0.371
40	0.680	0.886	0.656	1.037	0.623	0.812	0.411
45	0.700	0.946	0.668	1.048	0.625	0.845	0.452
50	0.723	1.014	0.680	1.063	0.626	0.878	0.491
55	0.750	1.096	0.693	1.083	0.626	0.915	0.531
60	0.782	1.197	0.708	1.105	0.626	0.958	0.572
65	0.820	1.318	0.723	1.135	0.624	1.002	0.613
70	0.873	1.464	0.745	1.172	0.624	1.045	0.644

A_6 and A_9, the upstream pressure taps should always be located between these points for the practical application of the orifice.

TABLE 13 ORIFICE COEFFICIENTS FOR 3-IN. PIPE

Ratio $r = \frac{d^2}{D^2}$	$(XC')_v$	$(XC')_p$	X	C'	C_v	C_p	$1-Z$
5	0.624	0.649	0.624	1.000	0.624	0.649	0.074
10	0.619	0.663	0.618	1.002	0.616	0.660	0.125
15	0.623	0.688	0.621	1.003	0.616	0.680	0.179
20	0.631	0.718	0.626	1.008	0.619	0.704	0.227
25	0.641	0.754	0.633	1.013	0.621	0.731	0.277
30	0.653	0.795	0.641	1.018	0.623	0.759	0.325
35	0.668	0.841	0.651	1.026	0.626	0.789	0.369
40	0.684	0.893	0.660	1.036	0.627	0.818	0.413
45	0.705	0.953	0.672	1.049	0.629	0.853	0.453
50	0.728	1.022	0.684	1.064	0.630	0.885	0.492
55	0.756	1.106	0.698	1.083	0.631	0.923	0.532
60	0.788	1.210	0.713	1.106	0.630	0.968	0.575
65	0.827	1.331	0.728	1.136	0.629	1.012	0.613
70	0.880	1.480	0.749	1.175	0.629	1.057	0.626

83 The curves in the upper left-hand section of Fig. 15 fulfil both of these conditions selected for good practice, that is, the upstream connections are always between 0.75 D and 2.00 D and the

downstream connections are at the vena contracta or at four and one-half diameters of the pipe. In this section are plotted as a function of the orifice ratio the values of XC', X, C', C and also the percentages of pressure restoration in the pipe. The values of these coefficients are given in Tables 8 to 13 for the various sizes of pipe.

84 The selection of the upstream and downstream distances to be used in general practice must not be taken as a limitation on the use of the orifice to these few points. With the proper care in

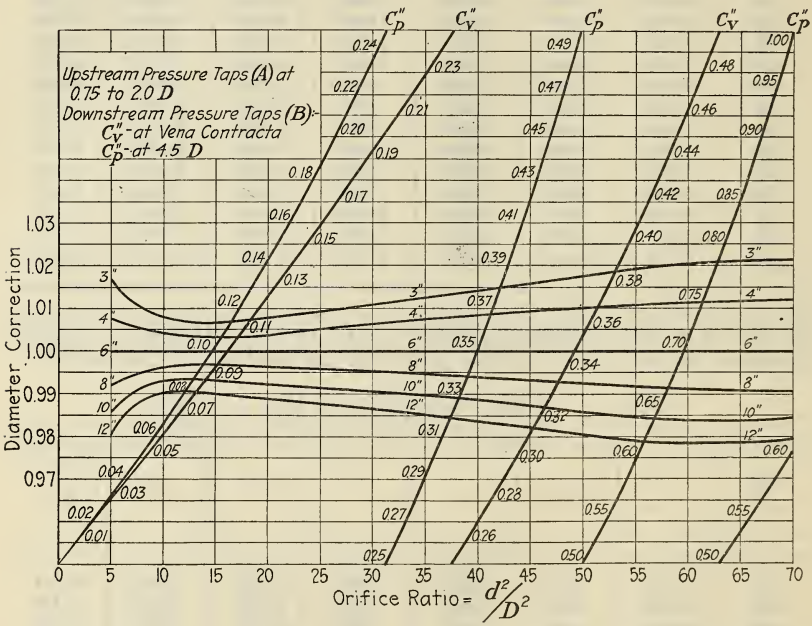


Fig. 16 Orifice Coefficient C'' for 6-in. Pipe and Correction Factors for Other Sizes

spacing the pressure taps, the coefficient corresponding to the exact distance can be easily and precisely obtained from the characteristic curve for the given pipe size.

85 In Par. 68 the orifice coefficient C'' was mentioned as affecting the simplest solution of orifice problems since it relates directly to the known section of the pipe D^2. Accordingly, Fig. 16 was prepared in a convenient form for ordinary use, giving values of C''_v and C''_p as a function of the ratio. To eliminate the necessity of repeating the curves for each size, only those for the 6-in. pipe were

prepared, and additional lines were drawn to scale giving correction factors to be used in obtaining the coefficient for the other pipe sizes.

86 Fig. 17 shows the spacing of pressure taps at the vena contracta in percentage of the diameter length corresponding to the given orifice ratio. The center line is a duplicate of the one passing through the minimum points of the characteristic *B*-curves in Fig. 15. The two outside curves, marked "min." and "max.," show the limits to which the tap may be shifted without varying the coefficients more than 0.25 per cent either way. The actual spacing should be made as nearly as possible to that given by the center

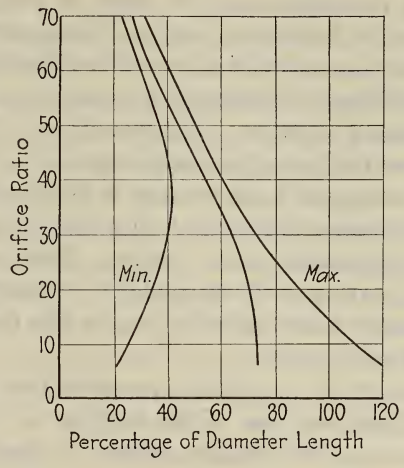


Fig. 17 Spacing of Taps at Vena Contracta

curve, since it is drawn to indicate the exact position of the vena contracta.

CONCLUSION

87 The scope of this paper will not permit of going into the particular details of each individual test. Its main object is to present the results obtained in the Research Laboratory in such a form that they can be applied conveniently.

88 A few remarks may be in order about the most salient points brought out by this investigation.

89 The orifice plate used as a measuring device for the flow of fluids in pipes has a relative accuracy well within the limit of 2 per cent. If reasonable care is applied in the selection of the

location, the proper spacing of the pressure taps, and the proper centering of the orifice between the flanges in the pipe, the possible deviation may be reduced to less than 1 per cent. The maximum variation from the average in any single test seldom exceeded 1 per cent when the differential pressure was above 1 in. of water.

90 The loss of head through the entire region affected by the orifice, which is the differential to the point of complete restoration, is an accurate function of the velocity squared — the same as that of the drop of pressure to the vena contracta. This means that the coefficient C_p is just as reliable as the coefficient C_v. In cases where the pressure difference has an appreciable value, it is even better to use the point of restoration, as the curve is much flatter at that point, in the case of large-ratio orifices, and a deviation from the proper section will not result in any appreciable error.

91 The coefficient of discharge C_v is shown to be very nearly constant for orifices of all ratios. This would mean that the orifice can be treated as a venturi, using the same value of C_v and correcting for initial velocity of approach the same as is done in the venturi tube.

92 The jet issuing from the orifice does not contract to the same extent in the smaller sizes of pipes. This is probably due to the fact that the portion of the flow which is restricted by the orifice has a proportionally lower velocity due to the increased skin friction in smaller sizes of pipes.

93 The variation of velocity pressure from 100 to 300 per cent does not affect the value of the coefficient. This seems to indicate that the name "frictionless orifice" is not misleading. For this reason it is best to use very thin material for the orifice plate to eliminate the friction at the sides of the opening. If the material has to be heavier, it is advisable to bevel the *downstream* side of the opening.

94 The regained head increases very rapidly with the increase of orifice ratio, making the larger-ratio orifice more applicable to cases where the loss of pressure is an important item.

95 Reference is made to the fact that the characteristic curves given in this paper are based upon the assumption that the coefficient of discharge for a 10 per cent orifice in a 12-in. pipe is equal to 0.61. This assumption, however, is borne out by the results of previous experiments by many authorities and can be further substantiated by additional volumetric tests at this point. The experiments, as demonstrated in this work, have shown that the orifice plate is relatively accurate for measuring the flow of fluids for the

varying sizes of pipes and the various ratios of orifices at the proper spacing of the pressure taps. The characteristic curves given in the paper have a complementary scale with the basic value of 0.61 as unity, so that if further experiments should prove that the 10 per cent orifice in a 12-in. pipe has a value somewhat different from the one given, these curves can be applied in the same manner to the new value of the standard.

DISCUSSION

R. J. S. Pigott. The paper on Orifice Coefficients is especially interesting in that the values obtained and the characteristics of the stream, when taken in conjunction with the dimensional analysis given in the Fluid Meters Report, afford a means of coördination of practically all of the experimental work on disc orifices. Heretofore these experiments have been thought to show considerable variation from each other, but when allowance is made for the difference of location in up-stream and down-stream taps, the ratio of orifice sizes, pipe sizes and the location of vena contracta, it is found that nearly all the experiments give coefficients substantially in agreement. Allowing for the difference of view of the various experimenters, it is very encouraging that this should be the case, and it is curious that the mathematical analysis of this flow prepared by Dr. Buckingham of the Bureau of Standards for the report should have predicted the characteristics of the orifice as found by Mr. Spitzglass, each working entirely without knowledge of the others' results. It is unfortunate that nearly every experimenter on both venturi tubes and the disc orifices should have used a different system of plotting and comparison; but possibly by trying all the different ways the best method will be adopted by the public who have to use them.

J. W. Hogg. The paper is very complete and covers the measurement of low-pressure air with the concentric straight-edge, thin-plate orifice for all ratios of areas up to 70 per cent in 2-, 3-, 4-, 6-, 8-, 10- and 12-in. pipes. One of its main features is the determination of two coefficients for a given ratio of areas in a given pipe depending upon whether the low-pressure connection is located at the vena contracta or at the point of restoration of pressure. The engineering profession certainly should feel indebted to the author for this extensive work, the results of which are now at its disposal.

The author states in Par. 95 that the 10 per cent concentric orifice in a 12-in. pipe was used as a basis for determining the curves shown on Fig. 15, and that a coefficient for this orifice of 0.61 was assumed as its true value which can be substantiated by actual test. Is the coefficient of this orifice 0.61 for water and steam? No mention is made of these two fluids. Barring the measurement of gas, the real commercial use of a flow meter is for the measurement of steam and water, particularly steam. It would, therefore, be well to know whether the coefficient of the basic orifice, as well as the results derived from its use as a means of calibration are applicable to steam and water, and possibly other fluids heavier or lighter than water, but possessing a different viscosity.

The writer believes that Fig. 17 covering the location of the vena contracta will be very serviceable. It shows the vena contracta to be sharply defined at high ratios of area and to spread out considerably as the orifice becomes smaller. Very often it is desired to measure a flow and an orifice ratio is figured for a given set of conditions, which include a guess at the maximum flow. A corresponding maximum differential is used (usually the range of the differential pressure instrument) to suit this maximum flow. After the orifice is inserted and the pressure connections made at the required distance, it may be found that the maximum flow is either greater or less, in which case it will be desirable to use an orifice with either a larger or smaller ratio. Provided the low-pressure connection was located exactly at the vena contracta, it will not necessarily follow that for the new ratio of areas it will be located at the vena contracta. The location of the low-pressure connection must be changed for this new installation. Fig. 17 shows that a good location for the low-pressure fitting is one-half of a pipe diameter behind the plate for ratios up to 50 per cent, therefore if the flow can be measured between this range without excessive differential head, a distance of one-half pipe diameter behind the orifice may be taken as a fixed location for the low-pressure connection. For the same reason, the fact that the low-pressure connection is at a fixed distance from the plate for the point of restoration of pressure, makes it also a very convenient form of measurement since its location will not have to be changed for varying conditions.

It may be noted on Fig. 14 that the coefficient of discharge for a given ratio of areas decreases as the pipe size increases. The author states in Par. 92 that the jet issuing from the orifice does not contract to the same extent in the smaller size of pipe as it does

in the larger, and he further states this is probably due to the fact that the portion of the flow which is restricted by the orifice has a proportionally lower velocity due to the increased skin friction in the small pipes. It then appears that the ratio of the perimeter of the pipe to the area of the pipe or some other similar function could be incorporated in a equation to be used in figuring the coefficients for sizes of pipe which are entirely too large to be erected in a laboratory for experimental purposes. After this equation is determined, the coefficient of discharge for these large pipes such as 18, 24 and 30 inches could be easily interpolated.

It may be also noted in Fig. 14 that the two sets of coefficients depending upon whether the vena contracta or the point of restoration are used as the location of the low pressure fitting are widely different in value, and the set of coefficients at the vena contracta may be termed as low coefficients, while the set of coefficients at the point of restoration may be termed as high coefficients. It appears from all the data taken in this test that sufficient is on hand to determine a set of coefficients with intermediate values with respect to the other two sets, and this set could be obtained by locating the high-pressure fitting at the point of restoration and the low-pressure fitting at the vena contracta.

Fig. 1, which shows the pressure gradient, is very interesting. The total hydrostatic head in the pipe is shown between the points A_6 and A_7 as $h + \frac{v^2}{2g}$ which is equal to the head with no flow, or equal to the head which would be shown by a pitot of unity coefficient. The pressure gradient curve from points A_5 to A_1 shows a decrease in the value of $\frac{v^2}{2g}$ and an increase in the value of h which is very peculiar. The writer believes that a possible explanation of this is as follows: Bernoulli's theorem states that the total hydrostatic head of a fluid under flow is equal to the static head at right angles to the flow plus the velocity head plus the friction head; or in symbols,

$$H = h + \frac{v^2}{2g} + h_f$$

Where H represents the total head, h the static head and h_f the friction head. Now at the point A_5 we have,

$$H_{A5} = h_{A5} + \frac{v_{A5}^2}{2g} + h_{A5f}$$

and at the point A_1 we have

$$H_{A1} = h_{A1} + \frac{v_{A1}^2}{2g} + h_{A1f}$$

Since the velocity at A_1 must be greater than the velocity of flow at A_5 from the law of continuity of flow, it follows that

$$\frac{v_{A1}^2}{2g} > \frac{v_{A5}^2}{2g}$$

Now since h_{A1} was found to be larger than h_{A5} by actual test,

$$h_{A1} + \frac{v_{A1}^2}{2g} > h_{A5} + \frac{v_{A5}^2}{2g}$$

If the friction head of flow remains about the same, namely: $h_{A5f} = h_{A1f}$ which might not be the case, it follows that $H_{A6} < H_{A5}$ to a greater extent than it would have been if the static head h_{A1} had decreased from h_{A5} in a ratio depending on the velocity head ratio. Perhaps the author knows the correct explanation.

A. F. Spitzglass. The author's evident intention in writing this paper has been to give to the profession a definite and precise knowledge of orifice coefficients and their physical meanings. In addition to its being a report of work done and the theoretical relations involved, the paper can be considered a complete and practical reference for orifice data. For that reason it may be well to elaborate somewhat on the application of this data to practical problems.

Fig. 16 shows a set of curves giving values of coefficients for a 6-in. pipe, and correction factors for pipe sizes varying from 3 to 12-in. This form was originally adopted to obviate the necessity of using a different curve for each pipe size. It was soon found, however, that these curves were unsuitable for determining the proper size of orifice by Equation [**11a**], Par. 68. The following example brings this out clearly.

Equation [**11a**] gives us

$$Q = C'' \frac{\pi}{4} D^2 \sqrt{2gh}$$

For a given pipe size, maximum flow and head we obtain the desired coefficient. We then apply this value to the curve to get the proper ratio. In the case of the 6-in. pipe we may read the value of the ratio directly from the main curve since the pipe size correction factor is unity. For the other pipe sizes, however, we must first

find a ratio corresponding to a value somewhat smaller than the actual coefficient. After several trials, a value may be found such that when multiplied by the pipe-size correction for the same ratio, it will give the desired coefficient, and thus we get the proper orifice ratio.

It seems best, therefore, for practical work in calculating orifices, that separate curves be used for each size of pipe. The values for determining these curves could be obtained from Tables 8 to 13 by multiplying the values given in the first column for (XC') by the orifice ratio (r).

Par. 91 mentions the similarity between the coefficients of discharge C_v as given in the equation

$$Q = C_v \frac{\pi}{4} \frac{d^2}{\sqrt{1 - r^2}} \sqrt{2gh}$$

with the coefficient of discharge C for the venturi tube. In Fig. 15 the value of C_v is shown to vary from about 0.61 to 0.62, or approximately 2 per cent. For smaller sizes the value of C_v shows much more variation.

It has been mentioned in this paragraph that since this coefficient remains nearly constant we can use some average value and find our constant for any ratio merely by correcting for velocity of approach as is done in the venturi. This statement must be qualified then as referring only to such cases in which a high degree of accuracy is not necessary. To obtain the true value of the coefficient from the value of C_v we must get this value exactly from a curve and then divide by $\sqrt{1 - r^2}$. Since this involves an extra step in the calculations, the coefficient C'' was adopted as the standard, and may be obtained directly from the curve for any ratio.

Since the completion of the tests the objection has been raised that the Ellison gages cannot be relied on to give results within the limits of accuracy necessary for laboratory work, unless calibrated individually against some absolute standard. Lest this cause reflection to be cast upon the value of the data given, it seems necessary to explain the methods adopted to overcome this difficulty.

From the description given previously it is evident that each test was essentially a comparison of two similar functions of the pressure head at two different points in the stream. Neglecting the corrections applied for change in area, specific gravity, etc., the coefficient varied as the square root of the ratio of the orifice differ-

ential to the head on the impact nozzle. Thus, only a relative accuracy between the two devices used for determining the pressures was necessary to give the required precision. With this in mind, tests were made on a large number of gages, checking them against each other over the complete range of their scales. Of these, three were finally selected that gave practically identical readings along the entire scale. One of these was used for obtaining the orifice differential and the other two for obtaining the head on the impact nozzle.

To retain this precision it was necessary, therefore, to choose a nozzle of such a cross-section that the two gages would read on the same part of the scale. Furthermore, in averaging the results of any of the tests, most weight was given to those runs in which the two pressures were nearly identical.

George S. Hendrickson. The calibration curves for the impact nozzles show that the coefficient changes when the velocity changes, due to the friction. From my understanding, the coefficient of the venturi tube is given as a constant irrespective of the velocity and the conclusion I draw is that both cannot be correct. Can anyone tell if coefficients of the venturi tube have shown a variation at different velocities?

All the tests were made for a given pipe size in the same pipe by varying the orifice ratio. In commercial pipe it is safe to assume that the inside surface of different pipes would not have the same characteristics. One piece might be very smooth and another quite rough. I believe that this might affect the orifice coefficient especially on the larger ratios and higher velocities.

C. A. Dawley. Mr. Spitzglass deserves the thanks of the Society and of the engineering profession for undertaking such a formidable series of tests as was required for this paper. The method of relative comparison between two orifices in series insures that the quantities measured are equal, because simultaneous, and this is the only practical method to use for such an extensive series of runs.

The primary reference standard is a 10-per cent (3.825 in. diameter) sharp-edged orifice in a thin plate, installed in a 12-in. pipe and assumed to have a coefficient of 61 per cent. The secondary standards are well rounded flow nozzles as shown in Fig. 6, with an impact tube mounted so as to give the impact head at the center of the flow nozzles. The arrangement of flow nozzles on manifold,

as shown in Fig. 3, is unfortunate from the fact that the air flow has to make three right-angle turns within a short distance, thereby creating violent disturbances or eddies in the flow stream. This piping would naturally produce swirling or helical flow in the vertical pipes leading to the flow nozzles, and would certainly give a higher velocity on one half of the opening of the nozzle than on the other half. The net result of both of these irregularities would be to give a low, and perhaps a variable, coefficient for the nozzles. These conditions of installation probably account for the erratic results shown in Figs. 8 to 13. After reaching a head of 4 to 5 in. of water column the curves are nearly flat and the approximate coefficient read from the curves is as follows:

Nozzle diameter, in.	Coefficient, per cent
1	96.8
2	96.5
3	96.6
4 (No. 4)	96.5
4 (No. 5)	97.5

On the 2-in. nozzle there are apparently two families of points which would give values about 95.8 per cent for the lower and 97 per cent for the higher group. The coefficient of a nozzle of this type has an absolutely definite value and should be higher for the large diameters than for the smaller ones. The writer would expect the 4-in. nozzles to have a coefficient of about 99 per cent at a head of 4-in. water column, if installed under favorable conditions for smooth flow. The fact that the coefficients found do not increase systematically with the diameter and that they are uncertain or variable in some instances, supports the belief that they are influenced by the conditions of installation. It would be of value if Mr. Spitzglass would explore, with a search tube, the whole area of the nozzle outlets as installed, finding the velocity across two diameters at different radii compared with that at the center, and then that the 3- and 4-in. nozzles should be installed in turn directly at the end of the 12-in. pipe in which the disc orifice is located and a further determination made on the relative coefficients.

It difficult is to compare series orifices of widely different size. The head created by flow of the same volume through two similar orifices in series varies inversely as the fourth power (approximately) of the diameter; for example a 1-in. orifice would give about 256 times as much differential as a 4-in. orifice. The writer would

suggest the following method of obtaining coefficients for flow nozzles to be used as primary standards for the disc orifices. Make four 1-in. nozzles numbered 1 *A*, *B*, *C* and *D;* four 2-in. nozzles numbered 2 *A*, *B*, *C* and *D;* two 3-in. nozzles numbered 3 *A* and 3 *B*, and one 4-in. nozzle numbered 4 *A*.

Standardize one of these nozzles, say number 1 *A*, by repeated tests against an absolute volumetric displacement. Then connect numbers 1 *A* and 1 *B* in series and take comparative readings which will show the difference, if any, between these two and by comparison with the absolute tests on number 1 *A*, will give an exact coefficient for number 1 *B*. Similarly with the other 1-in. nozzles. Then put all four 1-in. nozzles in parallel and connect in series with them nozzle number 2 *A*. The drop across the four 1-in. nozzles will be about the same as across one 2-in. nozzle. This will enable finding the coefficient of number 2 *A* and in the same way the other 2-in. nozzles. Then try two of the 2-in. in parallel against each of the 3-in. nozzles and finally four of the 2-in. against the 4-in. nozzle. This permits building up by relative methods, the exact coefficients of nozzles of widely different capacity, at heads all of the same order, with a minimum number of absolute tests on one size.

The most important point settled by Mr. Spitzglass's paper is the location of the vena contracta and the point of maximum restoration of pressure beyond the disc orifice. For the range of heads covered by his tests, these results should stand regardless of any minor inaccuracy in the coefficients of the impact flow nozzles as found.

E. G. Bailey. The work represented by the data given in this paper is very extensive and should form a valuable contribution to the important subject of metering the flow of fluids. It is unfortunate, however, that the author has deemed it best to present the data in such form that it cannot well be correlated or compared with other data previously presented to the Society on a similar subject. I refer particularly to the paper by Horace Judd [1] submitted at the Spring Meeting, 1916, on Experiments on Water Flow through Pipe Orifices.

In Professor Judd's paper as well as in other works published on this subject the orifice ratio, which is a very important factor in comparing results, has been used as $\frac{d}{D}$ while Mr. Spitzglass

[1] See *Trans.* vol. 38, p. 331.

prefers to use $\frac{d^2}{D^2}$. This makes it necessary for anyone desiring to compare results to recalculate and replot all curves.

It is also very unfortunate that Mr. Spitzglass did not have a standard or basic measurement upon which to start the comparison. The flow nozzles were calibrated on an assumed coefficient of one orifice, then the orifices were recalibrated on the basis of the flow nozzles. The same flow nozzle could not be used for all the different sizes of pipe so that the final results are rather far from a definite absolute basis. I believe the results as a whole give coefficients too high. I also believe the paper shows more difference between the coefficients of different sizes of pipe than actually exists. While most of the data may be within 2 or 3 per cent of correct absolute value, I think in the present state of the art greater accuracy is expected of the meter manufacturer. It is hoped that some subsequent tests can be made to substantiate or modify these absolute values as mentioned by Mr. Spitzglass in the last sentence of his paper.

Referring to Fig. 1, the right-hand view does not show the true stream lines on the inlet side of the orifice. The particles next to the wall of the pipe continue until they strike the orifice plate and then come in at right angles. The slight building up of pressure near the inlet side of the orifice is due to the reaction as the flowing fluid is diverted towards the center of the pipe. This right angle motion is also responsible for the vena contracta, the path of the stream on the outlet side of the orifice following the resultant of the two substantially right angled forces. At a distance in from the outer circumference of the pipe on the inlet side of the orifice the stream curves somewhat as shown. This, I think, explains the phenomenon of the pressure increase better than is given in Mr. Spitzglass's paper.

It would be interesting to know why Mr. Spitzglass has limited his test to a 70 per cent ratio orifice which is equivalent to 83.70 diameter ratio.

The means for measuring the differential pressure has evidently been confined to Ellison gages. No mention is made of calibrating them. I think it is necessary to do so as slight variations in the diameter and straightness of the glass tube introduce very appreciable errors in this class of work. Such calibration should go to some basic standard as merely checking for zero reading and leveling is not sufficient. Perhaps errors due to this

cause have been eliminated by smoothing out the curves when plotted.

The per cent "regained pressure" on the outlet side of the orifice as plotted in the upper left hand part of Fig. 15 and the location of connections at vena contracta as shown in Fig. 17 agree very closely with data submitted by the writer to members of the Research Committee on Fluid Meters some two years ago.

The use of circular concentric orifices gives no latitude as to the location of the vena contracta. This is rather annoying when high hub flanges are encountered; also when a pipe is tapped for an orifice of one capacity and later an orifice of a different capacity is to be installed. This may require retapping the connections at a comparatively short distance away so that it cannot be accomplished without going to another position on the circumference of the pipe. I have found it very advantageous in such conditions to use eccentric and segmental orifices as shown in Fig. 18.

It is interesting to see the attention being focused upon the thin plate orifice in pipe lines for the measurement of fluid. There is no question but what it has advantages over all other forms of primary devices as to accuracy and reliability as it is not so likely to become fouled up with scale, dirt and sediment as in the case of pitot tubes, venturi tubes and flow nozzles.

John L. Hodgson. The writer has read with interest Mr. Spitzglass's paper on Orifice Coefficients which repeats work which he briefly summarized in Fig. 18 of his paper on the Commercial Metering of Air, Gas and Steam which was read before the Institute of Civil Engineers in April, 1917.

He would like to offer the following comments and criticisms:

1 The contour of the impact nozzles shown in Fig. 6 is unsuitable for low-pressure work, as the effect of viscosity, which is most marked at low flows, alters the shape of the stream lines, and hence the distribution of velocity across the section. The variation of the coefficient of discharge with p_2/p_1 for such a nozzle is shown at I, Fig. 17 of the paper above referred to. It will be seen by comparing curves I and II of that figure that for values of p_2/p_1 which are above the critical value (i.e. when p_2/p_1 is less than about 0.527), the coefficient of discharge changes very little with p_2/p_1 (i.e. with velocity) even though there are great differences in the contours of nozzles I and II. This, however, is not the case for pressure ratios (and velocities) below the critical; hence a nozzle contour

which was suitable for Mr. De Baufre's purpose was entirely unsuitable for that of Mr. Spitzglass, where it was essential to work at values of p_2/p_1 which were very nearly unity.

2 It is stated in Mr. Spitzglass's paper that the point of

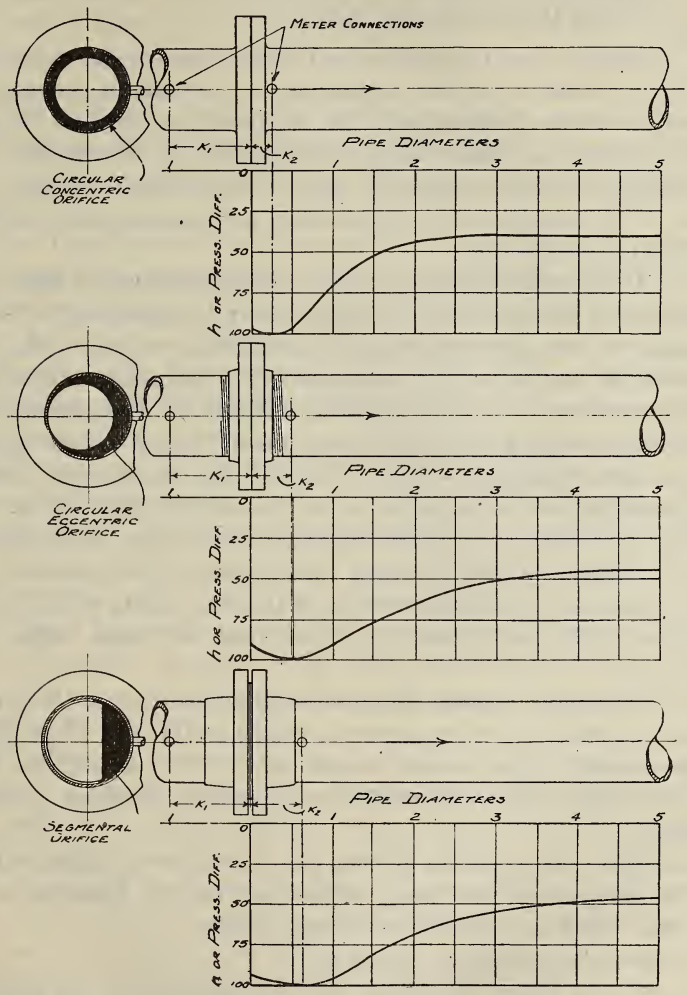


Fig. 18. Eccentric and Segmental Orifices

maximum recovery is 4.5 diameters on the downstream side of the orifice, and that this position is independent of the velocity and of the orifice area ratio. More extended tests covering a greater

range of velocities and pressure hole positions would have shown that the point of maximum recovery varies

a With the velocity of the air,
b With the roughness of the pipe, and
c With the orifice area ratio.

The proposal to use pressure holes further downstream rather than the vena contracta for flow measurement purposes is, according to the writer's tests suitable only for the most approximate measurements (unless the orifice has been calibrated in its own pipe line) as variations in the roughness, and — in the case of large orifice ratios — the diameter of the pipe, cause very considerable variations in the flow coefficient.

3 It is unsatisfactory to use circular orifices for area ratios greater than 50 per cent, as small errors in centering, or in the diameter of the pipe upstream or downstream of the orifice cause considerable errors in the measurement. Assuming then that it is only necessary to choose suitable positions for the pressure holes for orifice area ratio of less than 50 per cent, it will be seen on reference to Mr. Spitzglass's Fig. 17 that if the downstream pressure hole is drilled half a diameter on the downstream side of the orifice instead of in the best position shown in that curve, no appreciable error is made in the discharge coefficients of the orifices. The writer has used this simple rule to enable his clients to fix the best position of the downstream pressure hole for some eight years past.

4 For many reasons the most reliable position for the pressure holes is in the plane of the orifice as shown at III, and IV, in Fig. 17 of the writer's paper above referred to. It is a matter for regret that Mr. Spitzglass determined no pressures at these particular points.

5 The writer thinks it may be of interest to point out that all the tests could have been carried out on one diameter of pipe had the VDW/μ relation been used, where

V is the velocity of the air,
D is a linear dimension, say the diameter of the orifice,
W is the density of the air, and
μ is the absolute viscosity of the air.

Had the orifices been similar, or had the edges been bevelled off on the downstream sides so that the ratio thickness of edge to diameter of pipe was kept constant, only one set of tests would

have been necessary to determine the coefficients of discharge for orifices of any area ratio for any diameter of pipe.

The VWD/μ relation does not, of course, apply when p_2/p_1 differs appreciably from unity, nor could it be applied to a range of pipes where the relative roughness varied considerably. No appreciable error would, however, have been made in using it for the sizes of pipes and values of p_2/p_1 with which Mr. Spitzglass worked.

The Author. It is very gratifying indeed to note that the paper has aroused so much interest and discussion. I will endeavor to cover the various opinions as they were brought up by the members.

Mr. Pigott stated, with reference to the Fluid Meter Report, that nearly all the experiments on record gave substantially the same coefficients for the disc orifice, but he regrets that every experimenter should have used a different system of plotting and comparison. Mr. Pigott, being the chairman of the Fluid Meter Committee, is well familiar with the details of this work and he understands that the process of correlating and comparing the various experiments naturally belongs to the Fluid Meter Committee, or to the public who use the experiments, and not to the individual experimenters.

Mr. Hogg raised the question whether the coefficient 0.61 for the 10-per cent orifice in a 12-in. pipe is applicable for water and steam, as well as for air. The dimensional analysis given in the Fluid Meter Report shows that the values of the coefficient depend on the kinematic viscosity of the fluid, but from all experimental data available water, steam and the common gases show no difference in the coefficient of the orifice.

Mr. Hogg also suggested that a distance of one-half diameter should be adopted as the low-pressure connection of the orifice. This method has been used by many experimenters, but there is no advantage in adopting it for general practice. In a high-ratio orifice it is very important to place the connection at the proper point and it is best to adhere to this practice. If at any time the orifice is to be changed, it is not necessary to change the connection, as the difference in the factor can be easily obtained from the curves in the paper.

Mr. Hogg asked whether an equation could be derived to determine the effect of pipe size upon the coefficient of the orifice from the relation of the mean hydraulic radius as pertaining to

the friction drop of pressure. The dimensional equation referred to is supposed to correlate the results of tests so that the data of one size of pipe will apply to other sizes by following the law of dynamical similarity in the construction of the orifices.

Mr. Hogg suggested that an intermediate coefficient be determined by locating the high-pressure tap at the point of restoration and the low-pressure tap at the vena contracta. This is a very

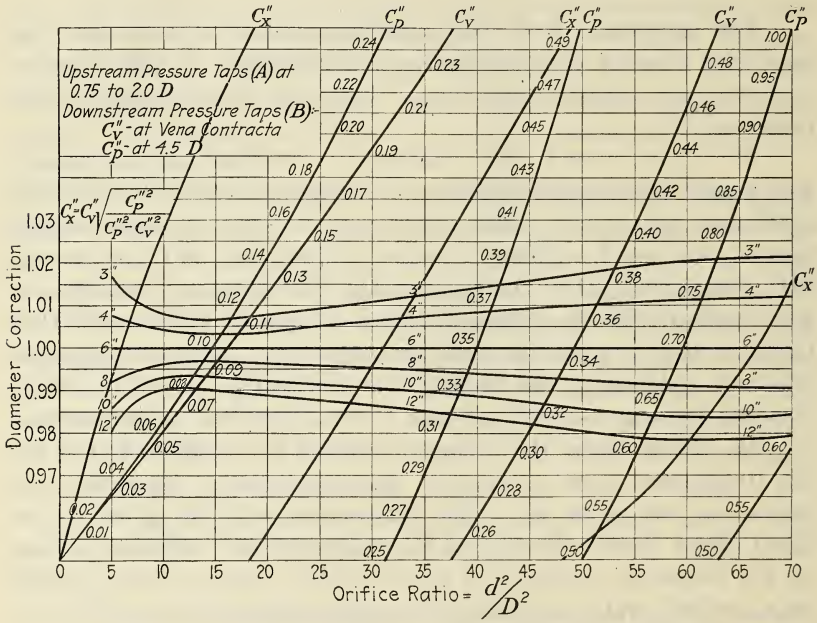


Fig. 19 Orifice Coefficient C''_x for 6-in Pipe and Correction Factor for Other Sizes

good suggestion and to take care of it, Fig. 19 is added, which is a modification of Fig. 16, showing in addition the values of

$$C''_x = C''_v \sqrt{\frac{C^2_p}{C^2_p - C^2_v}}$$

It will be noted, however, that the coefficient C''_x is intermediate only at high ratios, crossing the value of C''_p at 51 per cent, while at low ratios it is much higher than any of the other coefficients, so that the main advantage of this additional possibility would be increasing the capacity of the meter without the necessity of changing the orifice in the pipe.

Mr. Hogg was wondering how to adapt Bernoulli's theorem to the pressure gradient, shown in Fig. 1. He stated that from the appearance of the figure there is an increase in static pressure between the points A_5 and A_1, while from the nature of the flow there is also an increase in velocity between these two points, which is contrary to the theorem. As the fluid approaches the orifice the outer filaments are obstructed by the wall of the orifice and turned towards the center of the jet where the static pressure is the lowest, due to the increase of velocity pressure. This change of direction is the explanation of the vena contracta, which is the resultant path of the outer filaments of the jet. The static pressure shown at A_1 is slightly increased, due to the change of direction of the outer filaments of the flow.

The suggestions made by A. F. Spitzglass are a slight addition to the work he has done in preparing the results of this paper.

In reply to Mr. Hendrickson's inquiry, I wish to state that the coefficient of venturi tube does vary with the velocity of the flow similar to the variation shown on the impact tube curves. This variation is due to the effect of friction on the approach part of the venturi tube or nozzle. Since the coefficient of friction is generally increased at low velocity, there is a corresponding decrease in the coefficient of discharge, because the additional friction drop increases the relative differential pressure.

Mr. Hendrickson also inquired whether the roughness of the pipe affects the coefficient. From our experience the nature of the surface is not noticeable except at very high-ratio orifices. For that reason we prefer not to use orifices over the 70 per cent ratio of squares in which the effect of the surface becomes noticeable.

I was pleased to note that Mr. Dawley agrees with me on the method of relative comparison of orifices. As he states it was the only practical method to use for such an extensive series of tests.

Mr. Dawley objected to the arrangement of the flow nozzles on the manifold which followed a number of right angle bends. If Fig. 3 is examined carefully it will be noted that the first two turns are really long radius bends and of a size considerably larger than the approach of the flow nozzle. The air flow through any of the pipes loses practically all its velocity when entering the second manifold, which is of 14 in. diameter. The entrance to the third manifold is a long radius bend of the same 14-in. size. On the top of this manifold each riser is over 6 ft. long and the diameter of the riser is about one-half the diameter of the manifold, and is

double the diameter of the flow nozzle. So that, with this construction of approach there should be no disturbance or eddies in the flow stream.

Mr. Dawley's suggestion to explore the whole area of the nozzle was carried out during the test. We found that in case of the 4-in. nozzle, the impact pressure did not vary at all until the impact tube was placed $\frac{1}{4}$ in. from the circumference of the nozzle. At that point it varied about 1 per cent of the total impact and when the impact tube was placed so that the edge of the tube was in line with the edge of the nozzle, the reduction of the impact pressure was only 3 per cent of the total. This variation was the same in either direction of the nozzle, whether in line or across the turn of the flow at the bottom of the riser.

I believe that Mr. Dawley's statement, that the coefficient of a nozzle of this type has an absolutely definite value, is somewhat exaggerated. I am referring to the paper by Wm. L. De Baufre,[1] Calibration of Nozzles for the Measurement of Air Flowing into a Vacuum, presented to the Society in December 1920. In Table 2, page 635, the last column gives what Mr. De Baufre calls the average "weight-flow efficiency" of the various nozzles, which corresponds to our coefficient of discharge. I wish to call special attention to the last three nozzles, Nos. 18, 19 and 20 of the respective diameters, $\frac{3}{4}$, $\frac{7}{8}$ and 1 in. The first has an average coefficient of 0.9361; the second has an average coefficient of 0.9758, while the third has an average coefficient of 0.9598, and since I have used the same design of nozzle we may expect a similar variation in the results.

Mr. Dawley suggests to calibrate a number of nozzles in series to be used as a standard for calibrating disc orifices. I do not understand how the impact nozzle could be used in series and still be open to the atmosphere. If Mr. Dawley infers to calibrate the impact nozzle inside the pipe and correct for the static pressure, it would involve the old troubles of the regular pitot tube and the calibration inside of the pipe could not be compared to the one open to the atmosphere.

Mr. Bailey objected to the method I used in the paper to plot the results of the test on the basis of diameter square instead of diameter ratio, which he says was the basis used in Mr. Judd's paper and in other works published on this subject. Referring to Figs. 18 and 20 of Judd's paper, it will be noted that Mr. Judd plotted the coefficient of discharge and the factor of correction for

[1] See *Trans.* vol. 42, p. 621.

velocity of approach on the basis of the ratio of orifice area to pipe area, and not on the ratio of orifice diameter to pipe diameter. If Mr. Judd chooses to call a $2\frac{1}{2}$-in. orifice in a 5-in. pipe a 50 per cent orifice, while the results obtained with this orifice are plotted on the 25 per cent line in his main curves, it does not entail any difficulty to compare my results with his, when I choose to call it a 25 per cent orifice when it really amounts to that figure in the actual comparison of results.

There is another very elaborate paper on this subject by Holbrook Gaskell, Jr., published in the *Proceedings* of Civil Engineers, London, Vol. 197, on The Diaphragm Method of Measuring the Velocity of Fluid Flow in Pipes, and throughout this paper all the results are based on the ratio of areas or diameters squared, and not on the ratio of diameters. It is rather unfortunate that this method is not adhered to by all the manufacturers of fluid meters, as it deprives the user of the advantage of visualizing the flow through the orifice as compared with the flow through the pipe. When I think of a 5-in. orifice in a 10-in. pipe, I see before me a jet of 25 circular inches, as compared with an extension of 100 circular inches, which is 25 per cent of the total area. I cannot imagine a jet as a diameter but as a diameter square.

Mr. Bailey further objected to the standard or basic measurement upon which the comparison is based. I believe he will agree with me that the coefficient of a 10-per cent orifice in a 12-in. pipe is very close to 0.61, no matter what standard will be used for all the different sizes of pipes. In some cases two and three nozzles were used for calibrating one orifice, but each nozzle was calibrated separately, and therefore, the results are based on the same adopted standard. It is a common error to confuse the absolute accuracy of a device with the probable accuracy of each individual test performed for the purpose of calibrating the given device.

This point of difference was brought out very forcibly by Mr. Alfred Cotton in his paper The Accuracy of Boiler Tests,[1] presented to the Society in May 1922. Any individual test may effect the variation in the results for many particulars not at all adherent to the device, and the reason we are justified in drawing conclusions from the average results of a large number of individual tests is because the probabilities are that the errors will vary in both directions. Additional tests are being carried on now with a large holder to substantiate the standard adopted in

[1] See Paper No. 1841, this volume.

the paper, and these will be reported as soon as they are completed.

When I wrote the paper, I was familiar with Mr. Bailey's explanation of the vena contracta, but I never agreed with him that the particles of the fluid close to the walls of the pipe are actually turning on the square when they reach the orifice plate. If a particle has any momentum at all when it strikes a plane in the direction of the flow, it will rebound 180 and not 90 deg. as Mr. Bailey would have it in his analysis. The pressure gradient, shown in Fig. 1, was drawn from the actual tests recorded on log sheets 574–575. The jet outlined in the profile of the pipe was meant to show the boundary of the so-called region of disturbance, caused by the orifice plate. Inside that region there are numerous eddy-currents flowing in every direction, forwards, sideways and even backwards, and this is true on both the inlet and outlet sides of the orifice plate. The pressure gradient, as shown on Fig. 1, clearly indicates the extent of that disturbance and the fact that the pressure increases gradually from A_5 to A_1 is a further proof that the direction of the flow is beginning to change at A_5 and not at the very plane of the orifice.

With reference to higher ratio orifices, the tests were not limited to 70 per cent, but the results obtained for higher ratios indicated that extreme accuracy is required in centering the orifice plate and in measuring the pipe diameter. Higher ratios are being investigated at present and the results will be reported later.

The calibration of gages used for measuring the differential pressure is explained in the discussion by A. F. Spitzglass.

Mr. Hodgson's objection to the contour of the impact nozzle shown in Fig. 6 does not apply to this case, because in my tests the nozzle was only a means of conveying a stream of fluid and measuring the velocity pressure with an impact tube faced against the stream. This case is entirely different from the coefficient of discharge referred to in Fig. 17 of his paper where the velocity pressure was represented by the difference between the inlet and the outlet of the given nozzle. In my case the nozzle was used similar to the method employed by Mr. Sanford A. Moss [1] in his paper, The Impact Tube, presented to the Society in December 1916. In fact the preliminary tests referred to in my paper were made with a nozzle tube, similar to the one shown in Fig. 1 of Mr. Moss' paper. In the final tests the design adopted by Mr. De Baufre was used to see whether the change in design made any difference.

[1] See *Trans.* vol. 38, p. 761.

In both cases the same velocity pressure was obtained throughout the full area of the nozzle, as I explained in my reply to Mr. Dawley. There was only a change of 3 per cent in the velocity head, which would mean approximately 1.5 per cent velocity at the very circumference of the jet. This proved conclusively that the form of rounding the nozzle is immaterial when used in connection with an impact tube for obtaining the velocity pressure.

Mr. Hodgson states that the point of restoration or maximum recovery of pressure should vary

a With the velocity of the air,
b With the roughness of the pipe, and
c With the orifice area ratio.

In our tests the point of restoration did not vary when the velocity pressure was changed from 1 to 6 in. of water. Neither did the final point vary with the orifice ratio but the approach to the maximum recovery shows a different shape in the larger ratio orifice. The roughness of the pipe should have no effect on this point because the stream does not touch the walls of the pipes until it comes to this point. From this point farther on, the pressure drops, due to the effect of friction in the pipe. A 4.5-diameter ordinate on Fig. 18 in Mr. Hodgson's paper would cross the maximum with a slight exception of the two upper curves. The use of pressure taps at the point of restoration is recommended because the curve of the coefficient is much flatter at that point in the case of the high ratio orifices.

Mr. Hodgson does not recommend the use of orifices over 50-per cent ratio, for the reason that small errors in centering, or in the diameter of the pipe will cause considerable error in the measurement. It is a very good suggestion to emphasize the necessity of centering the orifice and of measuring the exact diameter of the pipe when larger ratio orifices are used. With the method of centering, shown in Fig. 6, and the correct measurement of the diameter of the pipe, the results appeared to be equally reliable up to an orifice ratio of 70 per cent. As stated in the reply to Mr. Bailey, higher ratios are being investigated at present and the results will be reported later.

Mr. Hodgson regrets that we did not take readings of the pressures in the plane of the orifice. This was not necessary, as by extending the numerous points in the first diameter length towards the plane of the orifice, the true pressure could be obtained at any

geometrical point of the given pipe. Similarly by extending the various curves on Fig. 15, you can obtain the coefficient for any geometrical point close or at the plane of the orifice. I can see no reason why the most reliable position for pressure taps is in the plane of the orifice. If the reason is, as Mr. Hodgson explains on page 28 of his paper, because the position of the pressure holes in different sizes of pipes should be geometrically similar, the same reason should apply to the vena contracta which must be geometrically similar for a thin plate orifice and besides the stream of the jet is more defined than at the plane of the orifice. According to our tests both the vena contracta and the point of restoration are geometrically similar on all the sizes of pipes we have tested, from two to twelve inches diameter.

Mr. Hodgson makes reference to the theory of dynamical similarity as applied to orifices in various sizes of pipes. Mr. Pigott referred to the same theory stating that Dr. Buckingham predicted the characteristics of the orifices and the behavior of the jet at the various geometrical distances as found by the test. Obviously many more tests will have to be run on various sizes of pipes before the validity of the geometrical similarity will be established.

No. 1871

REFINERY AND ROLLING MILL FOR MONEL METAL, HUNTINGTON, W. VA.

By W. L. Wotherspoon, New York, N. Y.
Member of the Society

This paper describes in detail the steps taken in selecting the site for the location of a mill for rolling monel metal. The economic problems involved are stated and the facts leading to their solution are given. The remainder of the paper deals with the layout of the plant, emphasizing some features of design that are of particular interest.

MONEL metal consists of 67 per cent nickel, 28 per cent copper, and 5 per cent other elements, and is a natural alloy. In appearance it resembles nickel and in tensile strength it is comparable with steel, while its resistance to corrosion is very high. Its resistance to acids together with its physical and working qualities (for it can be cut, rolled, welded, machined, and forged) makes a combination of properties which are not to be found in other common metals.

2 Monel metal is produced at the refinery in shot, pig, and ingot from monel-metal bessemer matte, consisting of approximately 56 per cent nickel, 24 per cent copper, and 20 per cent sulphur, the iron contents being only about 0.4 per cent. The mill product consists of forgings, merchant and sheet bars, wire rod in coils, and sheets.

3 The International Nickel Company, in order to provide facilities for the increased requirements of monel metal, has recently completed a refinery and rolling mill at Huntington, W. Va. The project may be considered as a new industry; active development of markets for monel metal having only been undertaken during the last few years, during which requirements have been met by arrangements with various steel mills for production of the

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above-mentioned products from ingots supplied from the Bayonne refinery of The International Nickel Company.

4 The purpose of this paper is to describe the investigation leading to the location of these works together with certain features of their designs and construction and to give information on the products manufactured, all of which is believed to be of general interest to engineers and manufacturers.

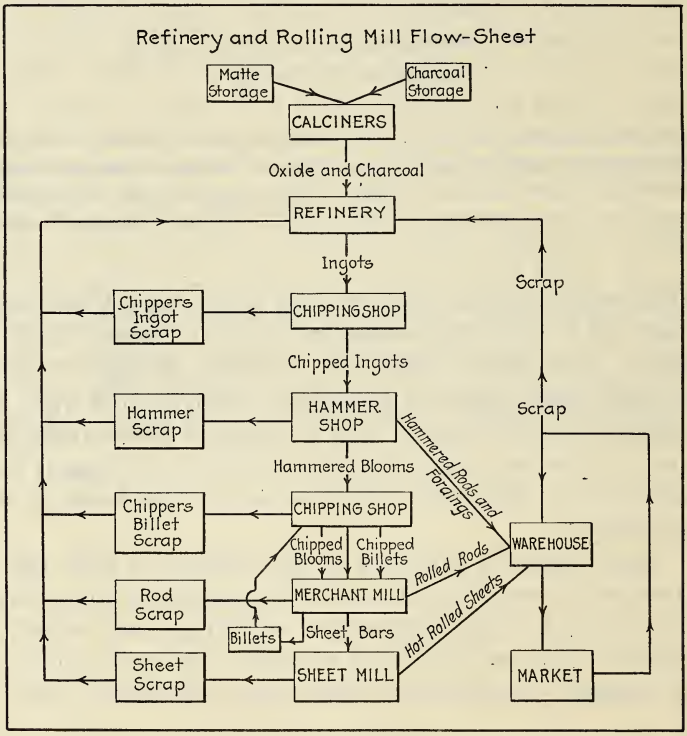

FIG. 1 FLOW SHEET OF REFINERY AND ROLLING MILL

SURVEY OF REQUIREMENTS

5 A general survey of requirements in order to prepare preliminary plans, and estimate the costs and acreage necessary, was first made, and involved the following:

a Analysis of past volume of business — sizes, forms and quality

b Estimate of potential markets

c Schedule showing minimum number of furnaces, mills and general equipment necessary to meet present requirements and those of the near future

d Flow sheet and general plans, Figs. 1 and 2, utilizing the above information and for the purpose of deciding on the number, size, and relative location of buildings and amount of equipment, in order that an estimate of construction costs may be prepared

e Estimate of operating costs which, together with capital requirements, is necessary in order to compare the cost of rolling monel metal by contract with steel mills of large tonnage capacity with that of a specialty mill of relatively small tonnage capacity

f Discussion of quality of product to be expected from specialty mill under the company's control, as compared with contract arrangements.

CONCLUSIONS OF SURVEY

6 As a result of the survey outlined the following conclusions were reached:

a That existing and prospective business for monel metal justified the estimated capital expenditure of approximately $3,000,000 for a rolling mill to produce rods and sheets

b That the potential markets for monel metal and other alloys were such that the plant should be laid out with provision for considerable expansion

c That natural gas, being practically free from sulphur, is the ideal fuel for heating purposes. Should it be necessary to utilize producer gas, the estimated capital expenditure for plant and equipment would be increased approximately by $200,000

d That the particular disposition and service of the merchant and sheet mills was such that individual electric drive, with gear reduction where necessary, would be best

e That purchased electric power was preferable, and if not available capital expenditure would be increased by approximately $750,000

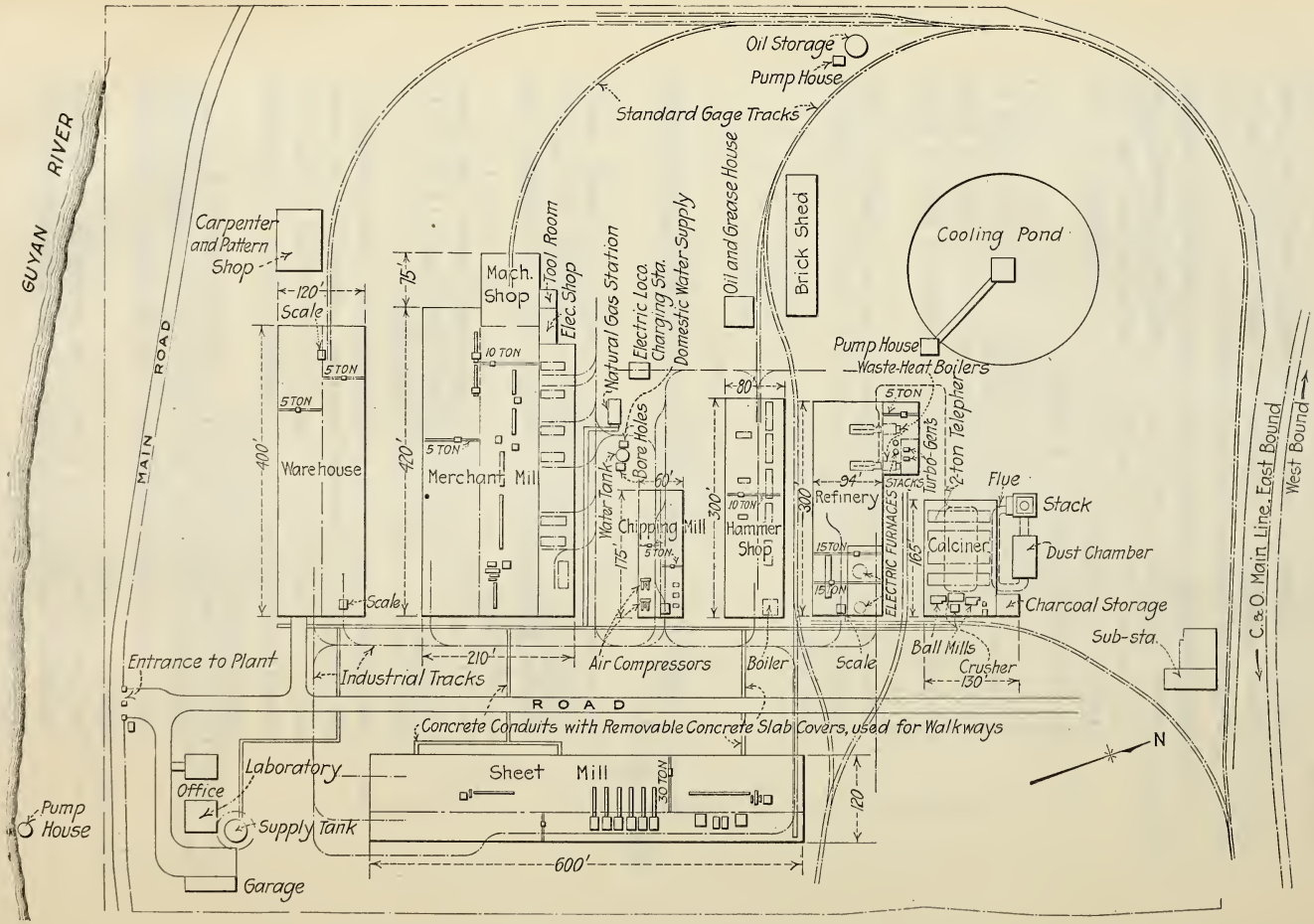

Fig. 2. Plan of Works, Showing General Arrangement

f That at least twenty, and preferably up to fifty, acres were advisable for a site.

INVESTIGATION REGARDING LOCATION FOR WORKS

7 With the survey of requirements and the conclusions available, the following districts were given particular study: Bayonne, N. J., Buffalo, N. Y., Baltimore, Md., Pittsburgh, Pa., and Huntington, W. Va.

8 Although the ores and bessemer matte from which monel metal is produced come from The International Nickel Company's mines and smelter in the Sudbury District of Ontario, Canada, and there had within a few years been constructed by the company a refinery for nickel products at Port Colborne, Ontario, economic factors such as tariff, fuel, markets for finished products, etc., were such as to confine the detailed study of locations for these new works to the eastern half of the United States.

9 Fig. 3 represents the territory and principal railroads which serve customers and prospective markets for refinery and mill products, including good facilities for export shipment. This territory covers the intensive manufacturing districts of the United States, with Chicago and St. Louis in the West, and New York and Philadelphia in the East, and it will be noted that Huntington, W. Va., occupies a geographic position central to the territory, and is also in close proximity to extensive natural resources, among which coal, oil and natural gas are the most important.

10 Bayonne, N. J., was considered, as The International Nickel Company's largest refinery had been established there many years, and it was thought the rolling mill might be an addition to the existing works. The investigation resulted, however, in the Bayonne refinery being discontinued and the plant dismantled. Important changes and extensions have therefore been made at the Port Colborne refinery so that nickel in all forms can be refined in Canada, and the refinery with rolling mill has been constructed at Huntington, W. Va., for the production of monel metal, malleable nickel, and other specialties.

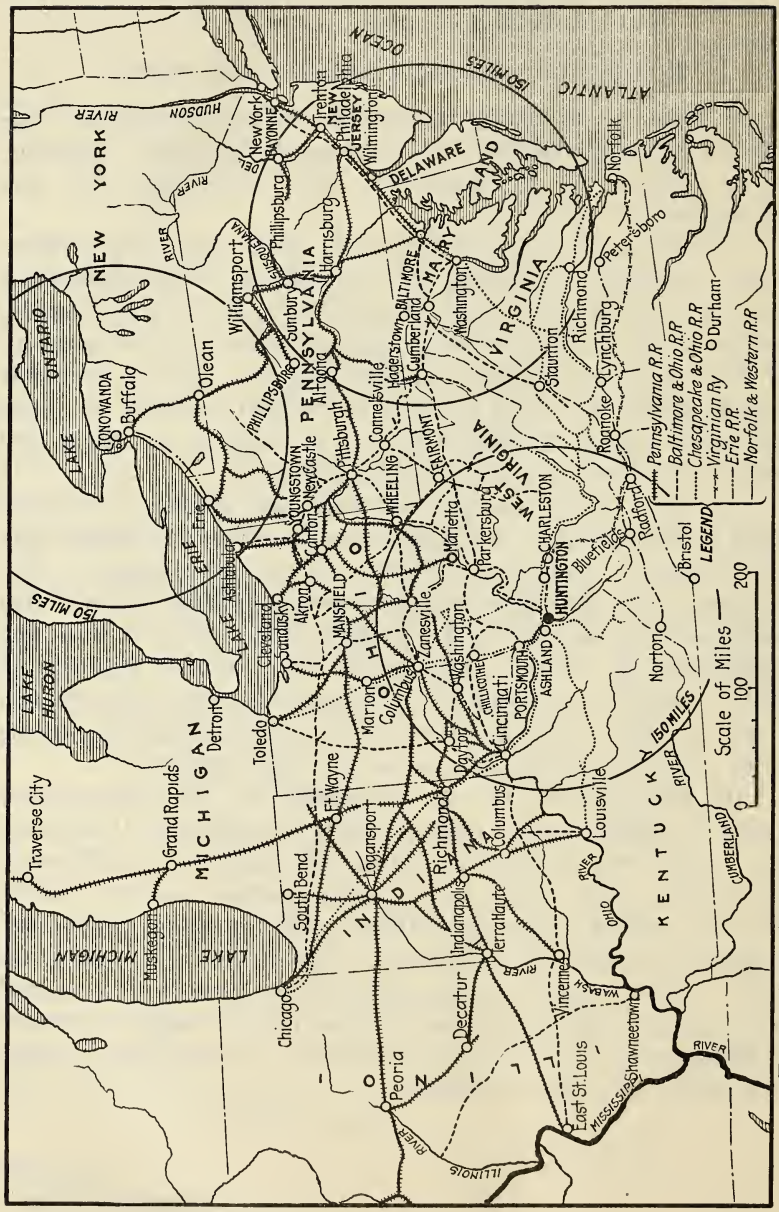

FIG. 3 MAP SHOWING RAILROAD AND MAJOR INDUSTRIAL PORTION OF THE UNITED STATES

11 The following is a schedule of economic factors investigated:

- Labor
  - Skilled
  - Common
    - Type
    - Supply
    - Rates
    - Strikes
- Fuels
  - Metallurgical
  - Power Generation
    - Cost and Quality
      - Oil
      - Producer gas
      - Natural gas
      - Coal
      - Coke
- Power
  - Public Service Electric Supply
    - Costs
    - Service
- Living Conditions
  - Housing
  - Cost of Living
  - Sanitation and Health
- Climate
  - Minimum and Maximum Average Temperatures
  - Average Snowfall
  - Average Rainfall
- Transportation
  - Railroads
  - Water
    - Sources and Costs of Supplies
      - Matte
      - Refractories
      - Rolls, Castings and Mill Spares
      - Sheet Bars
      - Charcoal
      - Electrodes
      - Chemical and Metallurgical — miscellaneous supplies
      - Lubricating Oils
      - General Stores
    - Products, Distribution of
      - Monel and Nickel Shot, Pig, Sheet Rod, Wire Rod, Forging
        - Domestic
        - Export
- Water Supply
  - Service
  - Costs
  - Quality
- Taxes and Laws
  - State
  - Local

Selection of Site....
- Railroad Connections
- Character of Ground for Building and Equipment Foundations
- Drainage and Flood Conditions
- Accessibility for Labor
- Grading and Facilities for Slag Disposal
- Provision for Expansion of Works

Construction Costs..
- Labor
- Material
- Supplies

12 The following represent the comparative ratings given to the economic factors:

Labor	250	Climate	50
Fuels	330	Supplies	60
Power	100	Taxes and laws	20
Living conditions	100	Site (cost and quality)	10
Transportation	50	Construction cost	20
Water supply	10		

13 The comparative ratings given economic factors for the principal locations studied are as follows, and were made during the latter portion of 1920:

	Locations				
	Huntington	*A*	*B*	*C*	*D*
Labor	100	90	82	75	65
Fuels	100	55	50	60	55
Power	100	90	95	90	60
Living conditions	100	90	90	85	80
Transportation	100	95	90	90	90
Water	100	90	75	85	15
Climate	100	90	85	90	85
Supplies	100	90	90	95	85
Taxes and laws	100	100	100	90	80
Site (cost and quality)	100	75	50	50	30
Construction cost	100	90	80	80	70

14 The following general conditions apply to the district of Huntington, W. Va., which are considered especially suitable for the industry described:

15 *Labor.* Labor is made up of 95 per cent English-speaking Americans, both common and skilled, with good records

in the territory in diversified industries; the turnover is light and a majority of the workers own their homes.

16 *Fuels.* A plentiful supply of natural gas for manufacturing and domestic purposes, from public-utility companies, is available at a cost of from 18 to 19 cents per 1000 cu. ft. (1100 B.t.u. per cu. ft.) Investigation of developed and undeveloped gas fields indicates supply of gas for 15 to 20 years. A good supply of high-grade oil of low sulphur content is available from the local oil refinery at a present price of 5 to 6 cents per gallon delivered at plant. There is an excellent supply of high-grade bituminous steam and gas coals from local coal fields costing $2.50 to $3.00 per ton, delivered at the plant.

17 *Power.* Two modern central stations supply power at a cost to large industries of 11 to 12 mills per kw-hr.

18 *Transportation.* Thirteen important railroads make connections within 175 miles. The Ohio River is navigable all the year, traffic between Pittsburgh, Huntington, and Cincinnati being on regular schedule.

19 *Water.* Both river and borehole water of good quality are available, the latter at an average depth of 60 ft.

20 *Climate.* The climate is equable, with generally cool nights and little snowfall.

21 *Supplies.* Refractories, charcoal, castings, and steel are obtainable in the district.

22 *Taxes and Laws.* These compare favorably with those of other districts. There are no smoke laws in West Virginia.

23 *Site.* A site of 76 acres was procured just outside the present city boundary of Huntington and directly connected to the Chesapeake & Ohio Railroad, and will be similarly connected with the Baltimore & Ohio Railroad when extensions now planned are completed. The site is also on the Guyan River and within about a mile of its junction with the Ohio River. The topography of about ten acres of the seventy-six is ideal for use as a gravity slag dump. Flood conditions in the Ohio Valley were studied and records for about forty years analyzed, which showed that only on two occasions — the last being the 1913 flood — has flood water been sufficiently high to endanger to a small extent property at the elevation of the site selected. The ground for foundations is excellent, being solid clay for a depth of from 12 to 18 ft., underneath which is a stratum of coarse, compact sand. The location was sufficiently close to Huntington, which is a growing city of

about 65,000 inhabitants, to obviate the necessity of the company building homes for officials or workmen. Housing conditions are not at this time very satisfactory, being similar to conditions in other prosperous communities, and rents and costs of houses will undoubtedly be reduced as building catches up with the requirements.

CRUSHING, GRINDING, AND CALCINING DEPARTMENTS

24 The matte, which is received from the company's smelter in Canada in irregular pieces weighing about 50 lb., is unloaded from box cars and put through a 24-in. by 16-in. jaw crusher, being reduced to about 1½-in. product, and is then fed to a No. 8 Krupp-type ball mill to be further reduced by dry grinding to pass a 16-mesh. The crusher and mill capacity is about 45 tons in eight hours, so that with the installation of a large storage bin with seven days' capacity for ground matte, this equipment can be operated intermittently to suit the convenience and arrival of cars of raw material. The crusher, ball mill, and elevators connecting them with each other and with the bin are belt-driven through a lineshaft by a 125-hp. 900-r.p.m. 3-phase high-torque squirrel-cage motor. There are two large flywheels on the crusher, and with regular loading of the ball mill, which is checked by a recording meter, this 125-hp. motor carries the load efficiently.

25 The ground matte is then introduced by mechanical feeders to roasting furnaces, of which there are three, the material being handled between the bins and the furnaces by 2-ton-capacity electric traveling monorail trolley hoists. The telpher has a hook lift of 30 ft., a hoisting speed of 20 ft. to 50 ft. per min., a traveling speed of 300 ft. to 350 ft. per min., and operates on a 12-in. I-beam connected directly to the lower chords of the roof trusses.

26 The roasting furnaces are horizontal, 80 ft. long by 17 ft. wide, and are of a modified Edwards type, the material being moved along the hearth by 32 rabbles operated by mechanical chain and gear transmission and mounted directly over each furnace. These rabbles are of cast iron with monel-metal shoes, and are hollow, being designed especially for efficient water cooling. Natural gas is used as fuel for these furnaces, the capacity of each being about 25,000 lb. per 24 hr., the sulphur in the material being reduced from 20 per cent to 0.005 per cent. The mechanism for each furnace is operated by a 5-hp. 220-volt 900-r.p.m. back-geared motor.

The temperature under which roasting is accomplished varies from 2000 deg. fahr. to 500 deg. fahr. The roasted matte is discharged from the furnace by gravity into large buckets, ground charcoal being mixed with the hot material as it is discharged so that its reduction takes place from the time it leaves the roasting furnace.

27 Large flues with numerous baffles, facilitating the recovery of valuable dust, were given special attention in these furnaces, these connecting with a chimney 200 ft. high and 8 ft. in internal diameter at the top.

28 Large quantities of charcoal are used, so in this department there is provided as an extension a brick building for the storage of charcoal. The charcoal is handled from railroad car to storage bin by a skip hoist driven by a 6-hp. d.c. motor with a Dinkey controller, which provides dynamic breaking. Charcoal is procured locally in irregular sticks and after passing over a conveyor provided with a magnetic pulley to extract any stray metallic substances is delivered to a gyratory crusher, driven by a 10-hp. 220-volt 900-r.p.m. motor, to be pulverized to about ½-in. size. A bin for the storage of crushed charcoal is provided, but the capacity is not large as it is inadvisable, due to the possibility of spontaneous combustion, to store this material in large quantities in ground form. There are also dangers of spontaneous combustion in storage in crude form, and bins of fireproof construction are subdivided to reduce the fire hazard.

29 The cooling of the furnace rabbles takes about 100 gal. of water per min., the temperature being raised to 160 deg. fahr., which heat is utilized by pumping the water to the boiler-feedwater storage system of the auxiliary power plant.

THE REFINERY

30 The refinery department contains two open-hearth furnaces arranged for firing by either natural gas or oil. The capacity of each furnace is approximately 35,000 lb. per day.

31 The electric traveling monorail-trolley hoist system already referred to operates between the calciner and the refinery departments, delivering the roasted monel-metal oxide from the calciner furnaces directly to large feed hoppers mounted over the open-hearth furnaces. The telpher also delivers to the same feed hoppers ground charcoal from the crushing and grinding department and amounting to approximately 25 per cent of the furnace

charge by weight. High-grade refractories are used in the construction of these furnaces owing to the high temperatures necessary, the tapping temperature being about 2822 deg. fahr. The heat utilized in furnaces of this type does not exceed 15 per cent of the value of the fuel, and a 600-hp. boiler of the waste-heat type, with superheater, is connected as close as practicable to each furnace, Fig. 4. The steam generated is used by the steam hammers

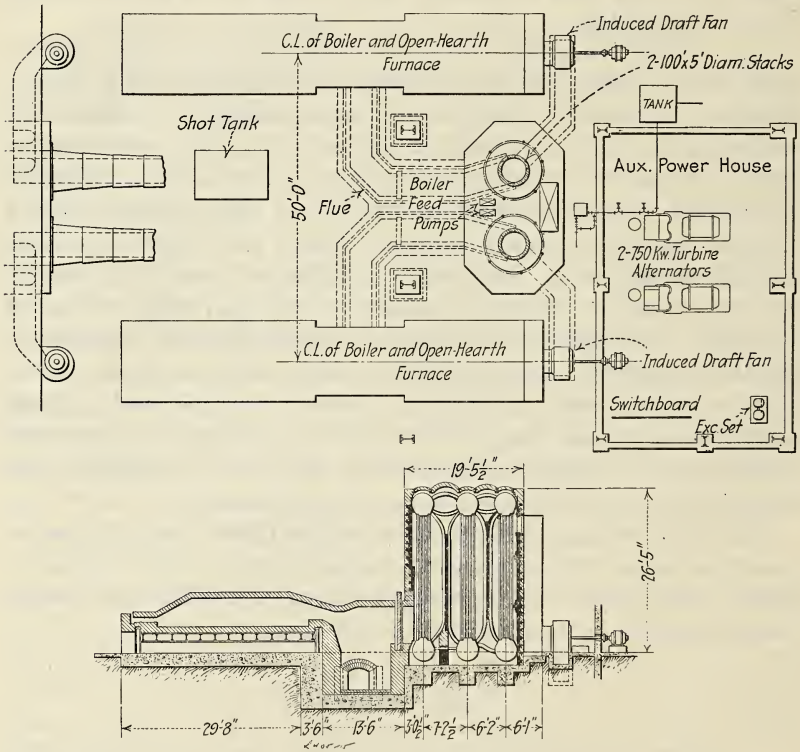

Fig. 4 Plan and Elevation Showing Arrangement of Open-Hearth Furnaces with Waste-Heat Boilers and Auxiliary Power Plant

and in the auxiliary power house, the latter being an extension of the refinery in which are installed two 750-kw. turbo-alternators.

32 The waste-heat boilers (6-drum type) were reconstructed from boilers of the 4-drum type formerly at the Bayonne Refinery, Fig. 5. The new installations have induced-draft fans of steel-plate construction with shafts of monel metal. Each fan is direct-

connected to a 30-hp. 600-r.p.m. a.c. motor. The auxiliary power generated will cost approximately 4 mills per kw-hr.

33 There are at the side of each open-hearth furnace specially designed tanks used when making monel metal in shot form. This

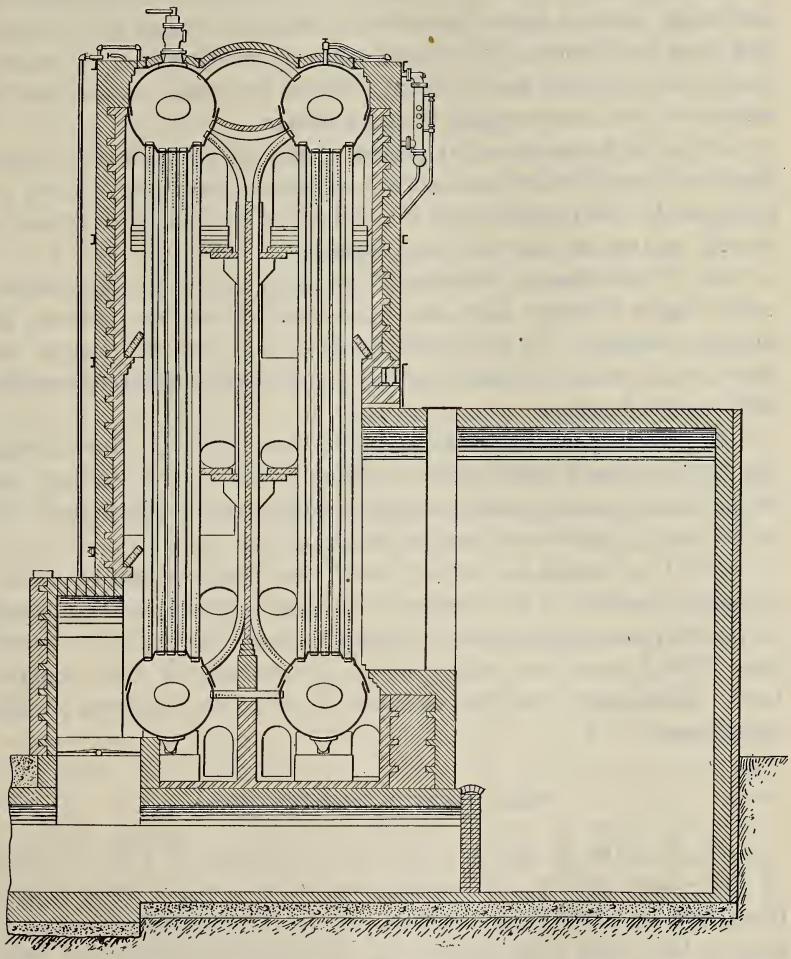

Fig. 5 Sectional Elevation of Waste-Heat Boilers Formerly Used at the Bayonne, N. J., Refinery

is accomplished by using a movable tapping spout and directing the stream of metal into water. There are also two electric furnaces, one known as the Moore, having a normal capacity of three tons per heat, and the other a Heroult, with a normal capacity of

seven tons per heat. These furnaces are installed side by side and operated from a common charge floor 12 ft. above the main floor, the space below being subdivided for transformers, auxiliary electric equipment, storage of electrodes, and general supply bins. The charge floor is 100 ft. by 35 ft., reinforced-concrete construction being decided upon after careful study of comparative designs and costs for structural-steel and concrete construction. The surface of the charge floor, where subject to rough usage, has an additional reinforcement of Irving grating.

34 The furnaces are at present basic-lined, and desulphurization and deoxidization are carried on under basic slag. They are arranged to tilt forward into the ladle and molding bay, which is served by two 15-ton traveling cranes.

35 The refinery department is equipped to produce monel-metal ingots directly from the open-hearth furnaces or from the electric furnaces. In the latter practice the furnace charge consists of pig metal produced in the open-hearth furnaces, together with selected scrap.

36 The operating practice with these electric furnaces has not yet developed sufficiently to afford authentic information, but in the work thus far done the power consumption varies from 642 to 714 kw-hr. per ton of melted monel metal.

37 The casting pit of reinforced-concrete construction is conveniently located to the furnaces, the ingots before being shipped to the chipping department having the heads removed by a cold saw. The floor of the refinery is of cast-iron brick on a concrete base. Equipment for the cleaning of slags is included in this department.

THE CHIPPING DEPARTMENT

38 In order to insure a uniform hammered or rolled product of first-class quality, the outer skin of the monel-metal ingots must first be completely removed. Previous practice has been to remove this outer skin entirely by chipping with pneumatic hammers, a costly and laborious operation.

39 Experiments on a small scale demonstrated the economic possibilities of milling the surface of the ingots, and three high-powered milling machines belt-driven from a 20-hp. motor were installed in the chipping department. During the few months of their use practical experiments have resulted in greatly improved

practice. Three additional milling machines recently ordered will embody certain improvements developed in the practice. One of the main problems encountered in the change from pneumatic chipping to milling of the ingots has been the question of economical handling and quick clamping of the ingots square with the table of the machine.

40 The ingot is placed on the table which is fitted with four points of hardened steel: one fixed, one adjustable, and two mounted on a rocker arm so that the ingot can be quickly trued up by the manipulation of the adjustable point. Two set screws are mounted on the bottom clamp casting for squaring the ingot with the table. To turn the ingot a fabricated-steel tower has been erected on one side of each machine supporting two chain slings so that with the table at extreme traverse the chains are placed one around each end of the ingot, the table lowered, and the operator with the aid of a U-bar easily turns the ingot 90 deg. for the new set-up. To overcome the tendency for the tapered ingot to slip longitudinally while being milled, and to eliminate the time wasted by the operator in tightening and loosening the bolts, together with the risk of damage to the milling cutters, a pneumatic clamp is now being designed which will decrease considerably the clamping and reclamping time, besides exerting a constant pressure and making it practically impossible for the ingot to slip.

41 Approximately 250 lb. of metal is milled from the large ingot in actual cutting time of 2 hr. 40 min. The total time to finish the ingot, including resetting of tools, turning and reclamping of ingots is 4 hr., thirteen operations being required. Eight-inch mills are used of special design with extra heavy steel bodies, case-hardened chip breakers, and ten inserted $1\frac{1}{4}$-in. by $\frac{3}{4}$-in. high-speed-steel blades. The blades are set in the body so that the rake angle which the face of blade makes with a radial line is 15 deg. Cutters of special design are used to mill the corners of the ingots, being mounted on a quick-change extension shank to allow for quick setting without removal of the face mill. The face mill is run at 16 r.p.m., taking $\frac{1}{4}$-in. depth of cut with feeds of $4\frac{1}{4}$ in. and $4\frac{5}{8}$ in. per. min. Angular cutters are run at 38 r.p.m. with $5\frac{5}{8}$ in. feed per min. The table of the machine with its load is raised and lowered at a proper vertical feed of 19 in. per min., and the horizontal adjustments are made at a rate of 150 in. per min. Considerable experimentation has been conducted in regard to proper cutter clearance angles and the following practice has been adopted

as the most satisfactory to date: namely, with the blades set in the cutter body at 15 deg. rake angle, to grind them as shown in Fig. 6.

42 In the early experiments an average of $3\frac{1}{2}$ ingots were milled before regrinding of cutter blades was found necessary. This average has gradually been improved and recently a high-speed-steel hardening gas furnace with preheating and final heating chambers was installed, together with lead pot, oil tempering and

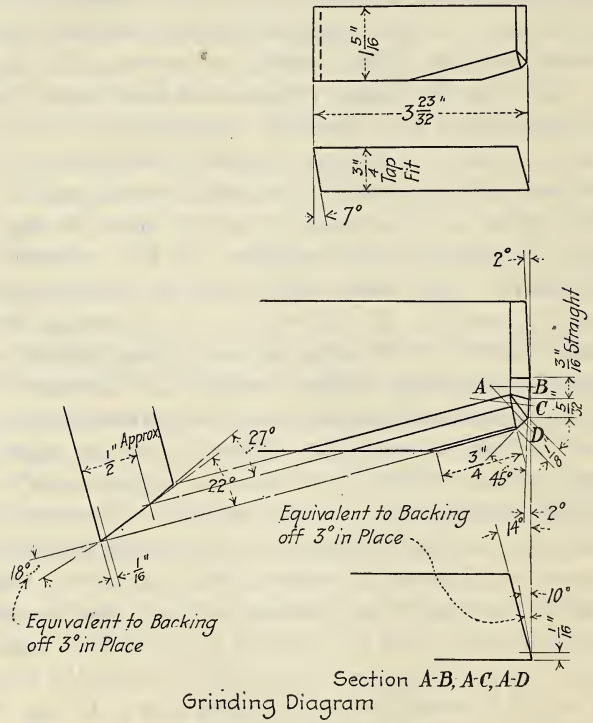

Fig. 6 Grinding Diagram of Cutters for Milling Machines

oil quenching tanks, and all temperature measurements are carefully made by a portable potentiometer using a chromel-alumel thermocouple. High-speed-steel blades hardened and drawn at 1150 deg. fahr. in this equipment have consistently cut on an average of 8 ingots with $5\frac{5}{8}$ in. feed per min. After cutting this number the tool just begins to dull and may be ground in a very short time with only a small loss of edge.

43 A skilled chipper cannot consistently chip out more than 1200 to 1500 sq. in. per eight-hour shift, or approximately two sides of a two-ton monel metal ingot, whereas one milling machine finishes an ingot every four hours. This time will be considerably lessened on the new machines with the improved devices referred to above. An average production from a battery of six machines of 12 ingots or 48,000 lb. per eight-hour shift may be consistently expected.

44 After the ingots have been milled they are taken to the chipping benches and carefully inspected. Any small defects that are not entirely eliminated by milling are chipped out by pneumatic chipping hammers.

THE AIR COMPRESSORS

45 Compressed air for chipping and miscellaneous work is supplied from two direct-connected electrically driven horizontal air compressors located in a corner of the chipping building where most of the air is used. This arrangement is also convenient for the service of one of the two 5-ton traveling cranes in this department. Aftercoolers were installed with the compressors because of the importance of having cool, dry air for pneumatic tools, and tests made have failed to find any moisture in the compressed air within a distance of 300 ft. from the air receivers. With such conditions the lubrication of pneumatic tools is also much improved. Fig. 7 illustrates the general arrangement of this equipment. The space is economically utilized and in practice the arrangement has proved satisfactory. Each compressor will deliver 1030 cu. ft. per min. at 100 lb. gage pressure, and is driven by a 210-b.hp. synchronous motor running at 257 r.p.m. The floor of the chipping department is wood block on concrete base.

THE HAMMER SHOP

46 This department receives the ingots from the chipping shop. The ingots vary in size, depending upon their ultimate use, the major portion being about 13 in. by 13 in. and weighing up to 4 tons.

47 The building is high, being 45 ft. from the floor to the bottom chord of the roof, due to the extreme height of the largest steam hammer, which is 28 ft. The building proportions give excep-

tionally good lighting and ventilation. The equipment consists of one 10-ton crane and four steam hammers of 16,000 lb., 10,000 lb., 3,500 lb. and 1,500 lb. capacity, respectively, together with the necessary handling equipment. There are five heating furnaces of the Stevens regenerative type, four with hearths 7 ft. by 20 ft. and one with a hearth 7 ft. by 14 ft., equipped with manual control and automatic air- and gas-regulating equipment. The re-

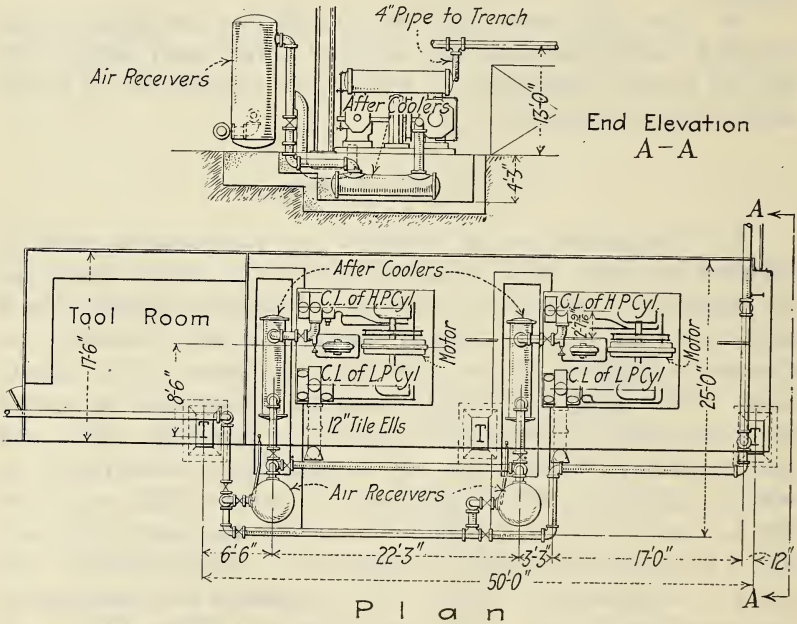

FIG. 7 PLAN AND ELEVATION OF AIR COMPRESSORS WITH AFTERCOOLERS AND RECEIVERS

generators are constructed at both ends of the furnace above the floor level.

48 Air for the heating furnaces is supplied at 16 oz. pressure by two steel-plate blowers, each of 7200 cu. ft. per min. capacity. These are driven by 50-hp. d.c. motors with variable-speed control giving a range of 1000 to 1700 r.p.m.

49 The boiler installation in the hammer shop consists of two 350-hp. units of the Stirling type, operating at 160 lb. pressure and located for the convenience of track connections, coal storage and ash-handling facilities, but at present piped for natural gas.

Under normal operating conditions the hammers, however, utilize steam from the waste-heat boilers in the refinery.

50 Owing to the weight and toughness of monel metal the hammers are of a special design and possess some novel features. The 16,000-lb. and 10,000-lb. hammers are used for cogging or breaking down. This is sometimes done on a mill or hydraulic press, but for this class of work a hammer has been found most suitable. These hammers are of a special double-frame type and are built of steel throughout. The main cylinders are bushed and valves of a new design assuring economy of steam and at the same time allowing for full control by the operator, have been embodied in their construction. The weight of one frame of the 16,000-lb. hammer is 44,000 lb., and the total weight, including the anvil, is 370,000 lb. The frames are held together at the bottom by a massive cast-steel base plate, giving a rigid construction. The ratio of the anvil weight to the falling weight is 15 to 1. Cushioned safety cylinder covers are used with these hammers, also special guides and shoes.

51 The two smaller hammers follow the same general lines except they are of the two-leg type and in some parts cast iron replaces steel. Special shapes are forged on these hammers, in addition to the usual forged work.

THE ROLLING MILLS

52 The mills consist of two departments:

a Merchant and wire-rod mills for the production of sheet bars, billets, rods of various sizes and shapes, and wire rod in coils

b A sheet mill for the production of hot- and cold-rolled sheets.

53 On account of the extremely tough character of monel metal and the tendency it has of cooling rapidly, it is necessary to have mills of great strength and rigidity. The type of material rolled is similar to alloy tool steel and it is very necessary to have mills with machine work and general finish the best of its kind and that are equipped with fittings that require a minimum of adjustment.

54 The hammered ingot or bloom after being overhauled to

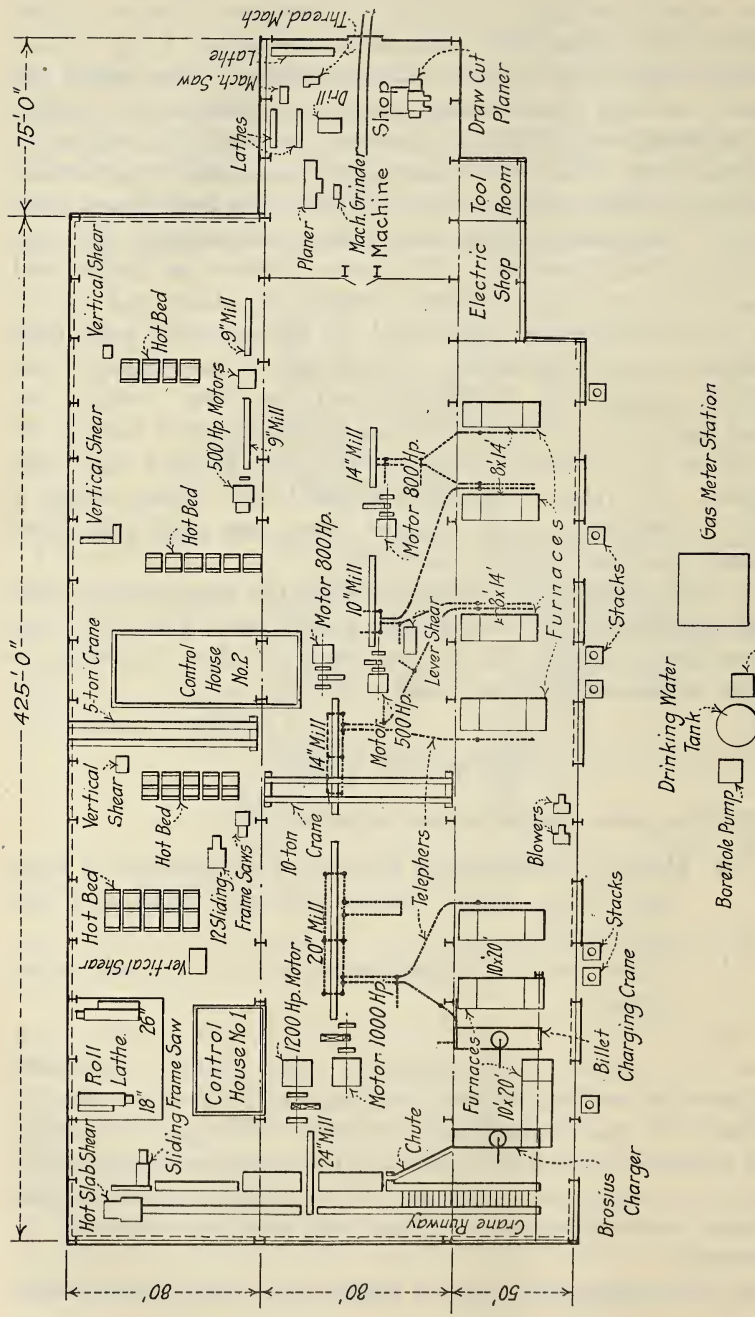

FIG. 8 GENERAL PLAN OF MERCHANT MILL

remove any scale or surface imperfections is delivered to the 24-in. sheet-bar mill, located in the merchant-mill building. This mill consists of two stands, the first stand three-high and the bull head, or finishing stand, two-high.

55 The handling of blooms to the heating furnaces and thence to the standard mill tables is by means of a Brosius charger, Fig. 9. The tilting tables, 25 ft. long, the transfer, and other table operations are controlled from the same pulpit. After being rolled the bar is carried to an exceptionally heavy shear with a capacity to cut $3\frac{1}{2}$-in. by twelve-in. billets.

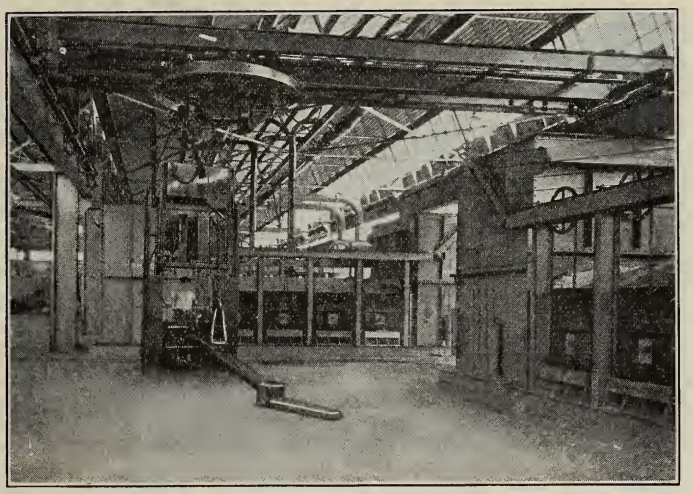

Fig. 9 Brosius Charger and Stevens Heating Furnaces in Merchant Mill

THE SHEET MILLS

56 The general arrangement of the sheet mills is shown in Fig. 10. The mills themselves are exceptionally heavy, the roll diameters being 30 in. and the maximum sheet width 48 in. The present complement consists of two finishing mills and two roughing mills, each finishing mill having a roughing mill of its own. The roughing rolls are balanced and are driven by fully enclosed cut gears, a rather new development in sheet-mill practice, giving a very smooth movement and reducing the tendency of the pinion teeth to mark the sheet. A specially designed drag, known as the "Conklin" drag, is furnished. The roughing rolls are screwed up and down by a motor drive, a 40-hp. motor being used which is

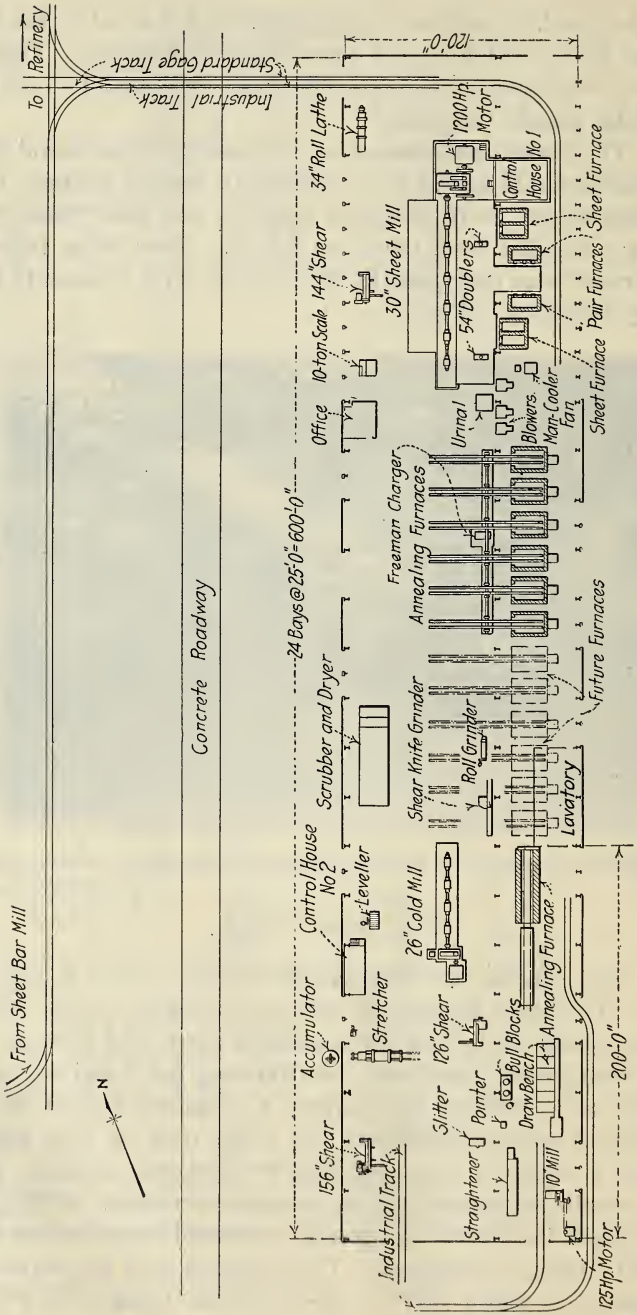

Fig. 10 General Plan of Sheet Mill

carried on the top of the pinion housing. The main motor to drive this mill is a 1200-hp. motor, which is geared with a double-reduction gear giving a total reduction of 13 to 1 and a mill speed of 26 r.p.m. Two 10-ton flywheels are used, running at the motor speed approximately 10,000 ft. per min.

57 The equipment is that usual for sheet mills, except for some slight changes in the floors. On the side nearest to the furnaces and in front of them the floors are of cast iron as usual, but are air-cooled for the roller and his helper. On the catcher side of the mill, however, wood blocks are used entirely except for a

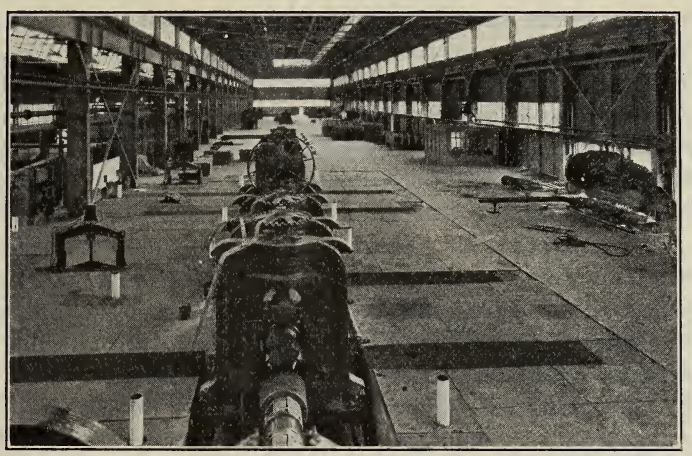

Fig. 11 Interior of Sheet Mill as Viewed from the North

small portion of grating floor, with air circulating below it at the catcher's stand. Doubling machines are furnished, operated by steam or air if necessity demands.

58 The furnaces are of the wide-door type, being double sheet furnaces and four-door pair furnaces. The air at 16 oz. is furnished by three blowers, of 4000 cu. ft. per. min. capacity, which also furnish the air for the six annealing furnaces. Natural gas is used in all furnaces. The annealing furnaces are equipped with Freeman-type charging machines. The crane capacity over this mill is 30 tons with a 25 per cent overload and a 10-ton auxiliary.

59 The 26-in. cold-roll mill at the south end of the building consists of four stands of cold rolls driven by a 300-hp. motor. Construction is standard throughout, except that the housings are very heavy, weighing 12 tons each. The arrangement permits an

addition of four mills on the other end of the motor drive. As there is very little fluctuation in the power required on this type of mill, no flywheel has been furnished, but a motor with large starting torque is provided. All of this machinery, including the gear sets, has given excellent satisfaction, and very little trouble has been experienced in starting up this mill.

THE MERCHANT MILLS

60 It will be noted from Fig. 8 that in the building with the 24-in. merchant mill there are other merchant mills used for furnishing the greatly varied product in tonnages that, from a rolling-mill point of view, are small. This equipment consists of the 24-in. mill referred to previously, which not only makes sheet bar but also the various sizes of billets required, a 20-in. mill, a 14-in. mill, a 10-in. mill, and a 14-in. Belgian rougher; and the wire mill, which consists of two separate mills in line with each other, 9-in. pitch diameter, one being a five-stand roughing mill and the other a four-stand finishing mill. These mills are equipped with gears and direct motor drive and all have a flywheel on the motor-shaft speed except the 9-in. roughing mill, which due to construction difficulties has the flywheel on the mill-shaft speed. The 9-in. finishing mill drives direct without any flywheel or gear reduction.

61 The cast-iron housings with cast-steel fittings and brass liners throughout are equipped with bottom screws arranged to be conveniently operated by the roller. With the exception of the two 9-in. finishing mills, the mills are what might be called slow-speed mills, as in spite of the fact that the material cools quickly, it is not advisable to roll it too fast, and this had to be given special consideration in deciding upon the speeds to employ.

62 All mills are driven by fixed-speed motors except the 10-in. mill, requiring a variation from 120 r.p.m. to about 80 r.p.m. to accommodate the varied size of material, and the 9-in. mill, the intermediate mill between the 14-in. roughing and 9-in. finishing mill, which will have a speed variation of 257 r.p.m. to 384 r.p.m., arranged in 17 intermediate steps, to accommodate the delivery speeds of the 14-in. mill.

63 The whole of the south side of this building is arranged with lifting doors, so that it is easy to adapt the temperature of this end to either summer or winter conditions, the same type of doors being arranged at intervals on the furnace lean-to side, as

will be noticed in the illustrations of the buildings. The lean-tos are arranged with an ascending slope away from the main building, so that the heat will travel away from the mill. This feature of special doors and lean-to has also been carried out in the equipment of the sheet-mill building.

64 Lubrication of mill reduction gears and pinions was given much study and a combined pressure and gravity system was installed in the control houses. Small plunger pumps are considered preferable for this service, as with their use there is not a tendency to form an emulsion as with some pumps of the rotary type.

THE BUILDINGS

65 The dimensions and arrangements of the buildings are shown in Fig. 2. All of the buildings, with the exception of the office, laboratory, oil and grease house and the electric substation, are of steel construction, the average weight per square foot of projected area being specified as 25 lb., although in some cases this is exceeded.

66 Wide monitors one-half the building width were adopted for the main building roof; the sawtooth roof was used for the lean-tos. These features give excellent ventilation and lighting. To assure the best working conditions during the warmer months, the prevailing winds were studied and the buildings were placed accordingly.

67 Steel lifting doors are continuous on the sides of the refinery, hammer shop, merchant-mill and sheet-mill buildings. For truck or railroad entrances to buildings, rolling steel doors are provided.

68 The roofing and sheeting is of corrugated black sheets; the windows have wooden sash, which take up any variation in the steelwork and eliminate warping of frames which otherwise might occur. There is in all about 160,000 sq. ft. of these windows used in the entire plant. The buildings were given two coats of a non-corrosive paint and a finishing coat of battleship gray color, which gives a pleasing appearance and aids the inside lighting.

69 In practically every important department suitable change houses have been provided. The washing facilities, showers and lavatories are arranged to utilize space efficiently, with a minimum of piping, valves, and fittings, and are sanitary and easily kept clean.

OIL AND GREASE HOUSE

70 The problem of the storage of oils and greases, together with a department for the rendering of grease used on the sheet mills, has been met by a small building of fireproof construction. The facilities for handling supplies are most convenient and Fig. 12 shows the general arrangement of this department.

TRANSPORTATION

71 Reference to Fig. 2 will show that every important department is served by track connections to the main line; also that suitable storage facilities for incoming and outgoing cars have been provided. All the standard-gage track is laid with 90-lb. rails, with large-radius curves, so that standard railroad engines may do the switching. The various departments are also served by a 36-in. industrial track, with electric storage-battery locomotives for the handling of materials.

WATER SUPPLY

72 Water was obtained from boreholes during the construction period and until the pumping station on the Guyan River was completed. The water it furnishes is of good quality, but it is chlorinated and tested daily as a precautionary measure, as it is used for drinking in addition to industrial purposes. The present installation (Fig. 13) consists of two 500-gal. DeLaval centrifugal pumps, each driven by a 40-hp. 2200-volt motor, operating against a head of 210 ft. and discharging to a 150,000-gal. tank.

73 In order to reduce the operating load on this station a pond of 10,000,000 gal. capacity, close to the auxiliary power plant, has been provided, and as subsidiary pumping-station connection with this pond supplies certain services of the plant; the water being in a closed circuit, the difference made up from the Guyan station is that due to leakage, which is very small, and the loss from evaporation.

ARRANGEMENTS FOR DESIGN AND CONSTRUCTION

74 No general contractor was engaged in connection with the construction of these works, principally on account of the special character of the work involved. The designs and specifications of

the buildings and mill equipment were prepared by Mr. Frank I. Ellis, consulting engineer, of Pittsburgh, Pa., and the designing of

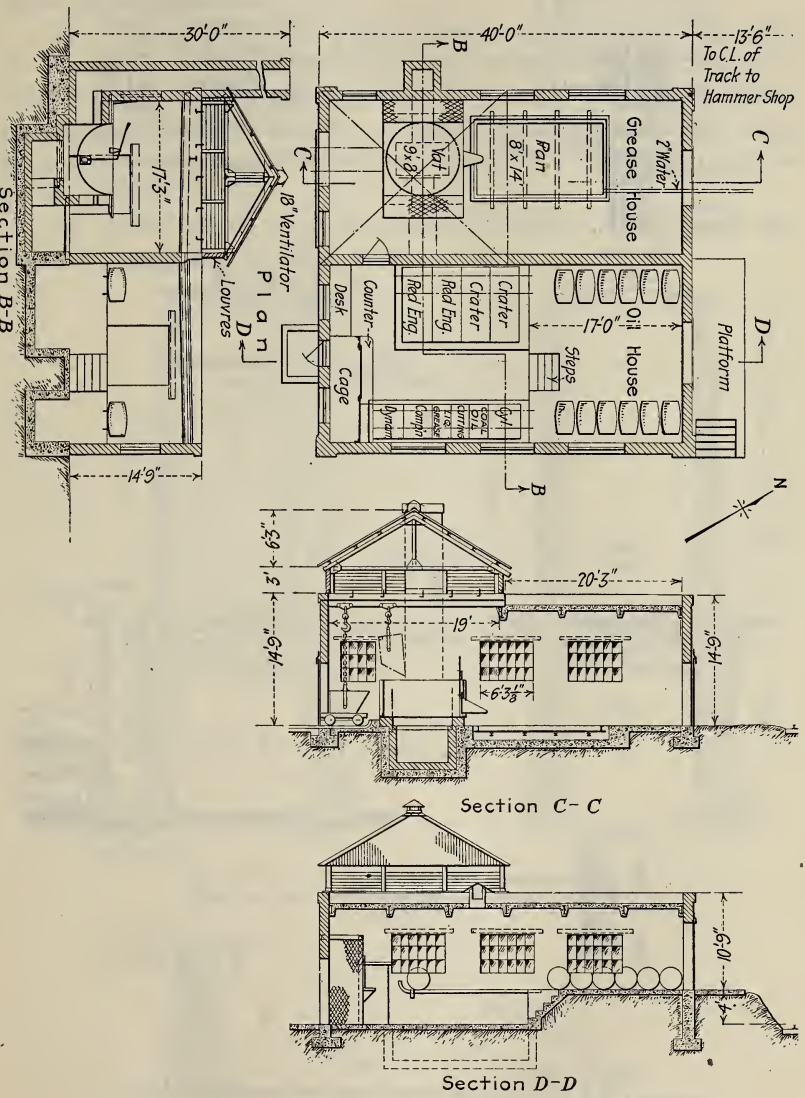

Fig. 12 Plan and Elevation of Oil and Grease House

the refinery, pumping station, and other details by the engineering department of The International Nickel Company at Huntington.

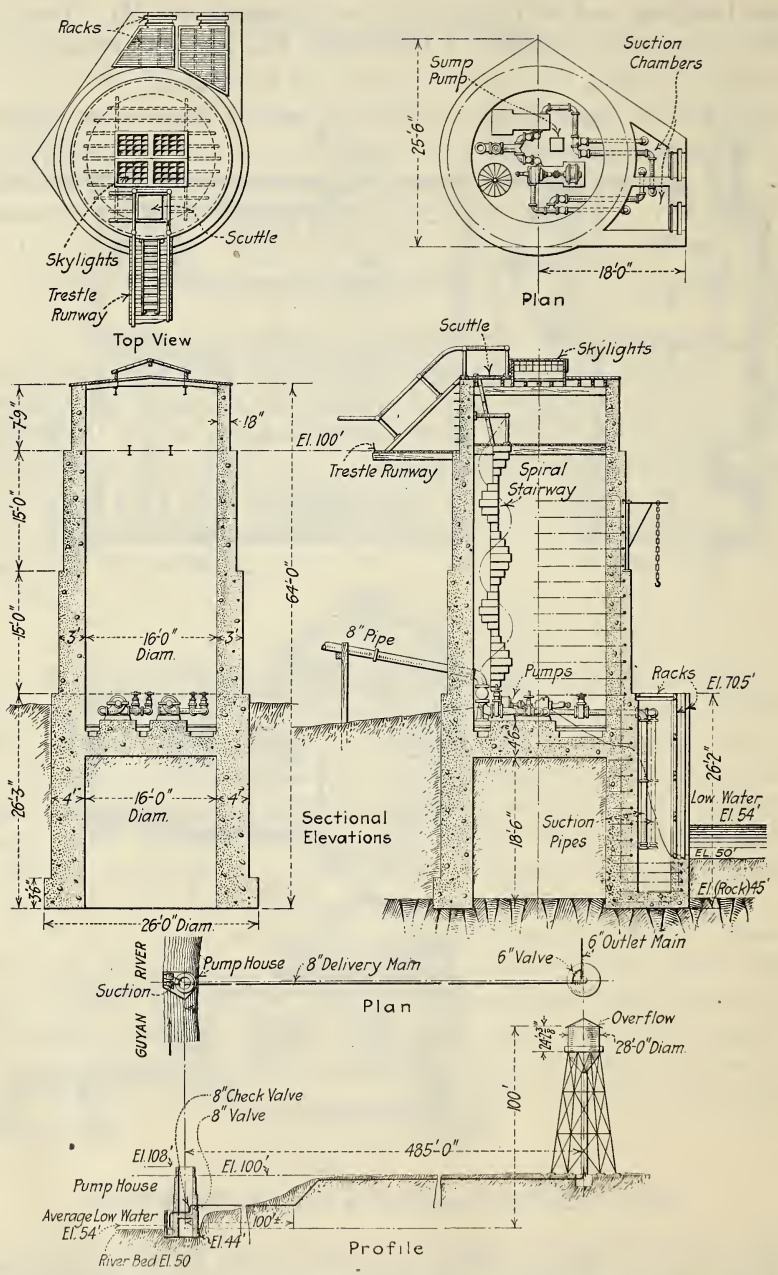

FIG. 13 PLAN AND ELEVATION OF PUMPING STATION (GUYAN RIVER)

Mr. H. M. Brown was engineer in charge of construction, and all the electrical equipment was installed by Mr. F. C. Watson, electrical superintendent.

75 The success of the undertaking is largely due to Dr. John F. Thompson, manager operating and technical department, The International Nickel Company, New York, and Mr. A. S. Shoffstall, manager, Huntington Works. All the engineering work, including the recommendation of the location and site, was under the direction of the author.

76 Construction was commenced in 1921 and the plant was placed in operation in June, 1922.

No. 1872

TESTS OF A TYPE W STIRLING BOILER AT THE CONNORS CREEK POWER HOUSE OF THE DETROIT EDISON COMPANY

By Paul W. Thompson, Detroit, Mich.
Associate Member of the Society

This paper deals with extensive tests conducted during 1921 *on a large Type "W" Stirling boiler at the Connors Creek power house of The Detroit Edison Company. Four different arrangements of the boiler baffling were employed and the results obtained with each are shown. Separate tests were made using four different grades of coal in order to determine which was most suitable. Several operating tests were made to obtain data on different methods of banking, the effect on the temperature of flue gas of varying the time interval between the blowing of flues, and different methods of operating boilers during the low-load period at night.*

From the results of the tests with different baffle arrangements, seventeen of these large boilers have been rebaffled with a resulting improvement in boiler-plant efficiency and an increase in the degree of superheat of the steam. Additional improvement is expected as more boilers are rebaffled. From the data and results of these tests, the Babcock & Wilcox Company, working in conjunction with The Detroit Edison Company, have developed a new design of this type of boiler. Four boilers of the new design have been installed in the Marysville power house of The Detroit Edison Company, but economy results on the installation are not yet available.

SINCE the original installation of the large Type W Stirling boilers by The Detroit Edison Company in 1911, many improvements have been made and much experience has been gained in regard to bettering the overall performance of these steam-generating units. The results of the first tests conducted on this type of boiler were presented before the Society by Dr. D. S. Jacobus in 1911. Several tests have been conducted since that time with other objects in view than to obtain accurately the efficiency, but none involving the weighing of water and coal.

2 Several years of experience in operating these boilers had

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers.

brought about many improvements, both in methods and in design of auxiliary equipment, yet it was felt that there still remained further opportunities for improving the operating performance. The determination of means for obtaining improved performance comprised the main purpose of these tests.

3 Eight series comprising fifty constant-rating tests were made, during which four different grades of coal were burned and four different arrangements of the boiler baffling tried out. Table 1 gives general data and dimensions of the equipment and Table 2

TABLE 1 GENERAL DATA AND DIMENSIONS OF EQUIPMENT

Boiler:

Type W Stirling, manufactured by Babcock & Wilcox Company. Total number of tubes, 1564; diameter, 3¼ in.; 9 gage. Net effective heating surface, 23,654 sq. ft. Installed and first in operation October 25, 1920.

Superheater:

Twin B. & W., U-tube. Two superheaters having 102 tubes each. Tubes 8 gage, 2 in. in diameter; developed length, 24 ft 11⅝ in. Total heating surface, 2996 sq. ft.

Stoker:

Underfeed stoker manufactured by American Engineering Company, Philadelphia, Pa. Two 13-retort, 2-ram stokers; total width, 22 ft. 10½ in.; effective length (both stokers), 12 ft. 5 in. (ashpit not considered in giving effective length). Projected grate area, 284 sq. ft. Width of ashpit, 3 ft. 11 in. Ratio of saturated heating surface to grate area, 83.4 to 1. Drive: variable-speed d.c. motors. Clinker grinders: independent drive by d.c. motors.

Setting:

Height of mud drums above floor	15 ft. 9 in.
Height from grate to top of combustion chamber	33 ft.
Width of combustion chamber	26 ft. 2 in.
Length of combustion chamber over grate	16 ft. 4 in.
Total volume computed above grate	8705 cu. ft.

Forced-Draft Fan:

Double conoidal blower manufactured by Buffalo Forge Co. Capacity, 74,000 cu. ft. at 6.5 in. water pressure. Driven by variable-speed d.c. motor.

Stack:

Height above center of grate	323 ft. 6 in.
Inside diameter of stack at top	15 ft. 9 in.

Dampers at boiler are motor-operated.

the names and sources of the coals, together with the quantities burned. Figs. 1, 2, 3, and 4 show respectively the four baffle arrangements: namely, the original baffle, the "A" baffle, the "B" baffle and the "C" baffle.

4 Some idea of the magnitude of the test may be gained from the fact that the total coal weighed to the stokers during the entire test was 12,137 tons, and the total water fed to the boilers 101,200 tons. During the highest-rating test the coal consumption averaged 22,550 lb. per hr. A complete record was maintained of the weights of coal and water during periods between tests, including seven

firing-up and burning-out periods, from which data an overall efficiency for the entire period up to and including test No. 76 was computed and found to be 77.9 per cent.

5 The tests were commenced April 7, 1921, and completed August 27, 1921. The full number of observations were taken until August 6 only, after which date operating tests were conducted for the purpose of determining the effects of blowing flues at various intervals of time and also to obtain data on the amount of coal required for different kinds of banks. These tests were conducted

TABLE 2 COAL DATA

(1) Name of Coal	(2) Location of Mine	(3) Tests	(4) Coal burned during tests of column (3), lb.	(5) Duration in hours of tests of column (3)	(6) Heating value of coal, B.t.u. per lb.
Shipper "A"	Harlan, Ky.	Nos. 3 to 18	6,785,888	613	—
Shipper "A"	Harlan, Ky.	Nos. 40 to 43	1,744,337	128	12,510
Shipper "A"	Harlan, Ky.	Nos. 60 to 66	2,857,053	242	—
Shipper "A"	Harlan, Ky.	Nos. 70 to 76	3,271,959	266	—
Shipper "B"	Mingo, W. Va.	Nos. 20 to 24	2,304,682	196	13,280
Shipper "C"	Kanawha, W. Va.	Nos. 18–30 to 33	1,668,998	143	12,320
Shipper "D"	Logan, W. Va.	Nos. 50 to 54	1,937,176	154	12,200

Averages for Whole Test:

Coal per hr., lb	11,780
Water per hr., lb	98,600
Per cent rating	145.6
Apparent evaporation	8.37
Equivalent evaporation	10.10
Per cent efficiency	77.9
Weighed refuse in per cent of coal burned	11.30
Computed refuse in per cent of coal burned	13.92
Loss to stack in per cent of refuse	18.8

with a reduced force, taking only the most essential observations. During the period from April 7 to August 6 the services of 57 observers working on three eight-hour shifts were required, besides those engaged in analyzing coal and ash, and others conducting special investigations and checking methods.

6 Three tanks were installed for the weighing of water, each tank having a normal capacity of 6500 lb. Two only were required to be in use at any given time, leaving one extra tank for use in an emergency or during the repairs or checking of scales. The receiving tank, which was located below the weighing tanks, had a capacity of 45,000 lb. of water and was of such cross-sectional area

that one inch in height gave a volume equivalent to 625 lb. of water. The water level in this tank, which was recorded each hour, was indicated at the weighing platform by a calibrated rod supported by a float in the tank. The feedwater was drawn from the receiving tank by a steam-turbine-driven boiler-feed pump which discharged

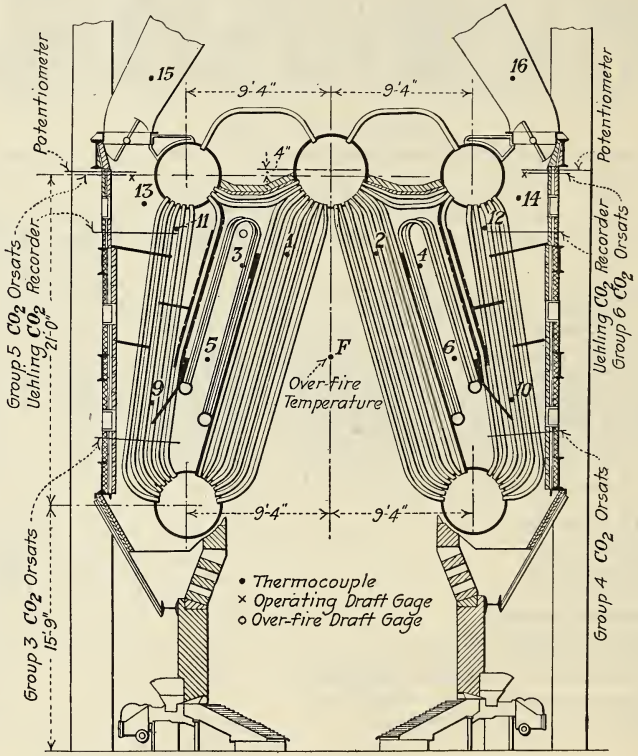

Fig. 1 Original Baffle Arrangement in the Type W Stirling Boiler Tested

only to the boiler being tested. Gland leakage from the pump was collected and measured.

7 All coal was weighed on the four scales provided, two of which were located on each side of the boiler. After being weighed the coal from each scale was dropped into a large funnel which delivered into the regular coal spreader.

8 The refuse was dumped from the ash hopper into standard steel cars and weighed at the end of each test on the track scale,

which is a part of the regular plant equipment used in weighing in all coal received at the plant.

9 A very elaborate scheme for taking simultaneous flue-gas samples from 24 points was devised and installed, but owing to the length of piping required to bring the samples to one point for

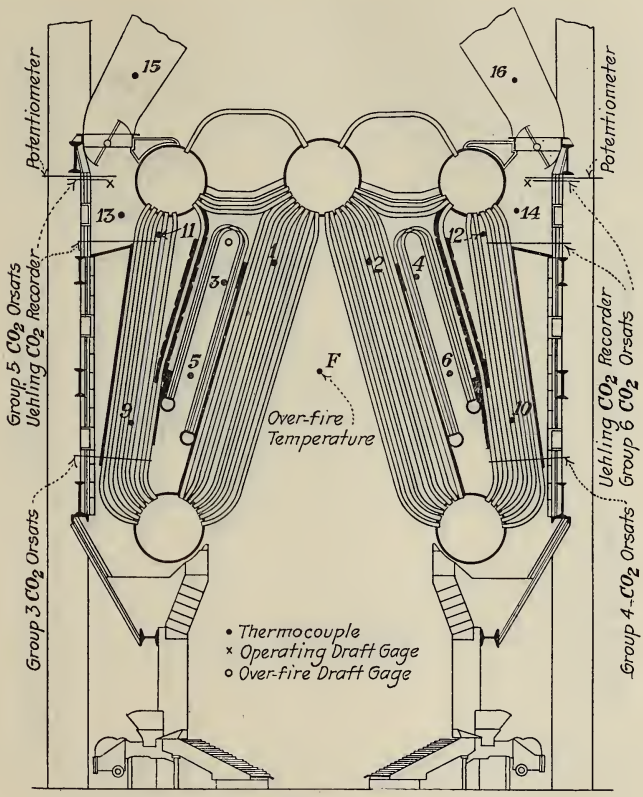

Fig. 2 "A" Baffle Arrangement

analysis, difficulty was experienced due to plugging of the lines. Further, it was realized that some error might result from the use of the long lines; therefore this method was discontinued after a few tests had been run. Facilities were then provided for analyzing the gas at a point as close as possible to the sampling point. There were two sampling groups on each side of the boiler, one at the bottom of the superheater pass and one at the damper. Fig. 5 shows a plan of the boiler with the arrangement of sampling tubes

at the damper. It will be seen that on the east side of the boiler, tubes *a* and *b* are placed in the south half, *c* in the center, and *d* in the middle of the north half of the space between the end walls of the boiler. This scheme of arrangement is reversed on the west side with *a* and *b* in the north half, *c* in the center, and *d* in the

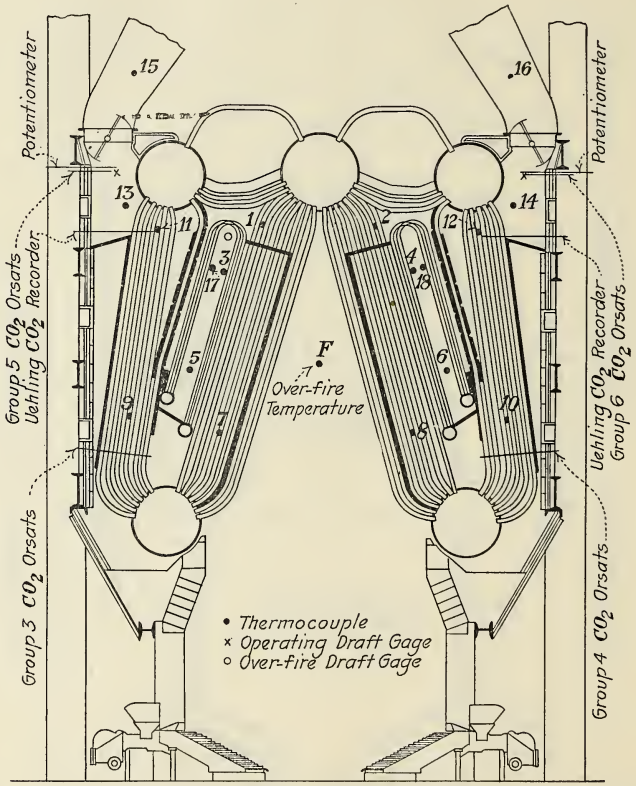

Fig. 3 "B" Baffle Arrangement

middle of the south half. The same scheme applies also to the location of sampling tubes at the bottom of the superheater passes on either side. This arrangement was chosen in order to check the composition of the gas in the diagonal corners of the boiler as well as on opposite sides. In addition to the points mentioned above at which gas samples were taken and analyzed in Orsats, two different types of automatic flue-gas recorders were installed, one drawing samples from four points on each side at the damper and recording an analy-

sis of an average of these samples, and another from one point on each side at the bottom of the superheater pass.

10 The points at which the pass temperatures were measured are shown on Figs. 1, 2, 3, and 4. All points on the west side have even numbers and those on the east side, odd numbers. Each

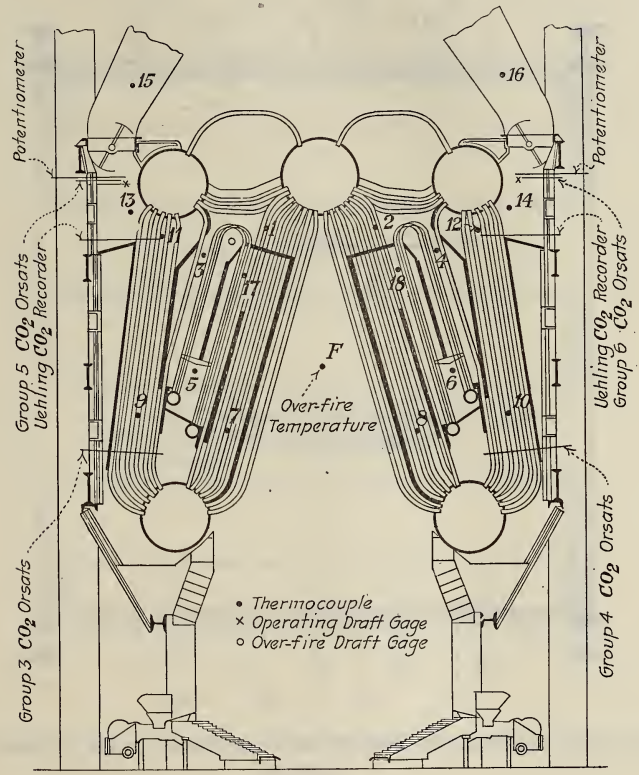

Fig. 4 "C" Baffle Arrangement

location except 15 and 16 represents two couples: one placed in the center C, the other placed in the south half S if it is on the east side, and in the north half N if on the west side. A plan view of the scheme of arrangement is shown in Fig. 6. Thermocouples made from iron-constantan wire were used for obtaining the temperatures at the points mentioned. The wires were carried through insulators protected, where necessary, by welded steel pipes to prevent the insulators from carbonizing. To reduce the radiation

effect, all hot junctions were shielded by means of sleeves made from 4-in. pipe and so arranged as not to obstruct the flow of gas over the couple. A portable potentiometer was used in obtaining the observations. Variations in cold-junction temperature were compensated by means of a nickel coil in the potentiometer circuit. The

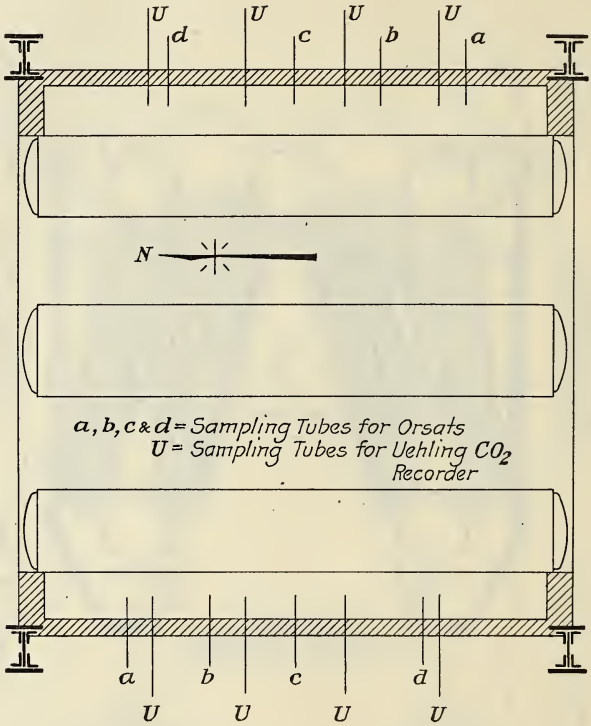

Fig. 5 Plan Showing Arrangement of Gas-Sampling Tubes at Damper

furnace temperature was obtained at point F at the north and south ends of the furnace by focusing an optical pyrometer on the closed end of a carborundum tube, the tube being of ordinary test-tube shape and projecting about 30 in. into the furnace. These temperatures do not represent the temperatures of combustion, since at the point of observation considerable heat had probably been lost by radiation to the first row of boiler tubes. In addition to the observations taken at the points mentioned above, a recording potentiometer was used for recording the temperatures at the

two dampers separately, each record being an average of five equally spaced points on either side as shown in Fig. 6.

11 In addition to the four operating draft gages regularly provided on the control board, the drafts were measured at points as close as possible to where thermocouples were installed.

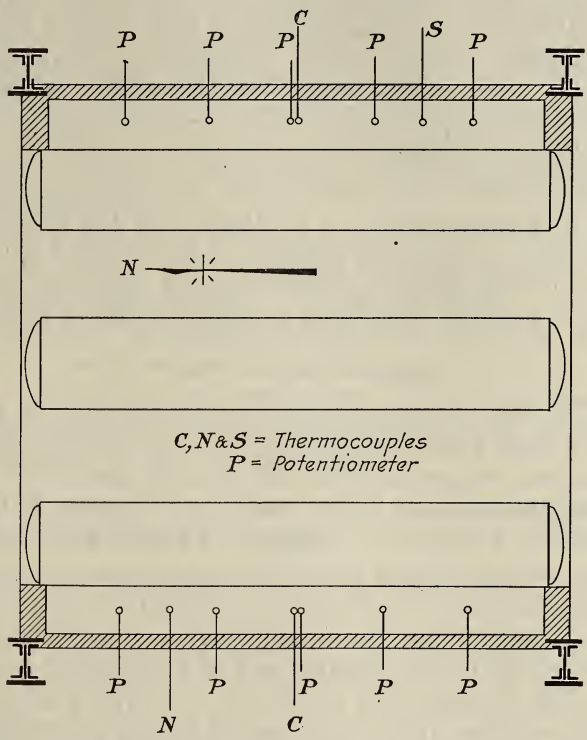

Fig. 6 Plan Showing Arrangement of Thermocouples

12 The quantity of air supplied to the air chamber beneath the stoker was measured by a Thomas air meter installed on the suction side of the blower. A housing was built over the blower and motor with an entrance duct having a cross-section of 6 ft. by 5 ft. and a length of 64 ft. to accommodate the metering equipment. The results obtained from the air meter differed from those obtained from the gas analysis due to the leakage of air from the air chamber. This difference increased with the rating, as would be expected, due to the higher air pressure, with a resulting greater leakage.

13 All testing equipment was installed and the tests were run

under the direct supervision of the author, assisted by Mr. A. K. Bak, technical engineer of the Connors Creek station. Coal and water-weighing equipment was installed by the Construction Engineering Bureau of The Detroit Edison Company. The Thomas air meter and thermocouples were made and installed by the Research Department under the personal supervision of Mr. W. A. Carter. Miscellaneous testing equipment such as draft piping, gas-sampling piping, electrical meters, flowmeters, thermometers, flue-gas recorders, etc., were installed by the regular operating and maintenance force at the plant. Each week throughout the test those particularly interested met for the purpose of discussing results and methods. The writer wishes to express his appreciation to Mr. J. W. Parker and Mr. C. F. Hirshfeld of The Detroit Edison Company, and to Dr. D. S. Jacobus of the Babcock and Wilcox Company for their valuable suggestions and criticisms, and to all those who in any way contributed to make the tests a success.

DESCRIPTION OF TESTS

14 The eight series of tests were made up of 50 constant-rating tests, each series being conducted under different conditions. The first series, comprising tests Nos. 2 to 10, was for the purpose of obtaining the efficiency before making any changes in the boiler or its auxiliary equipment. Shipper "A" coal was used since a regular and uniform supply could be counted upon, and its quality was about an average of the many kinds being received at that time.

15 At the time the tests were commenced the boiler had been in regular service for six months, and it was decided to ascertain how much the efficiency was impaired by the soot which gradually collects on the tubes and is not removed by the soot blowers. To this end the first series of tests was conducted without specially cleaning the boiler. Then the boiler was given a thorough washing on its exterior surface and a second series of tests was run, comprising tests Nos. 11 to 18, also using Shipper "A" coal.

16 When Dr. Jacobus made his tests on this type of boiler in 1911 he used Red Jacket coal, so for the purpose of comparing results obtained in 1911 with those obtainable now, a series of tests was made (Nos. 20 to 24) using Shipper "B" coal, which, in quality, was as close an approximation to the coal used by Dr. Jacobus as could be obtained. This coal, however, was of poorer quality, having a lower heating value and higher moisture and ash than the 1911 variety.

17 Tests Nos. 30 to 33 were made with Shipper "C" coal to determine its burning characteristics and adaptability for use on this type of stoker. The boiler baffling was then changed to conform to that shown in Fig. 2, called "A" baffle arrangement, and a fifth series of tests (Nos. 40 to 43) was made using Shipper "A" coal. Another series of tests using Shipper "D" coal was also made with this same baffle arrangement (tests Nos. 50 to 54) in order to obtain the burning characteristics and adaptability for use of this coal.

18 After test No. 54 the baffle was again changed to the "B" arrangement shown in Fig. 3, and a seventh series of tests (Nos. 60 to 66) was conducted to obtain data on the boiler operation with this baffle arrangement.

19 The last of the constant-rating tests (Nos. 70 to 76) were made with the boiler baffled according to arrangement "C" shown in Fig. 4, during which Shipper "A" coal was burned.

20 Following the constant-rating tests, additional tests were conducted to determine the effect of blowing the flues at different intervals of time upon the temperature of the gases leaving the boiler. Tests Nos. 80 to 83 were conducted for this purpose, with intervals between blowing flues as follows:

Number of test	80	81	82	83
Interval between blowing flues, hours	4	12	24	8

During these tests the boiler rating was maintained at about 155 per cent.

21 The following nine days the boiler was banked during the night and over Sundays to determine the amount of coal required for banking. For the first five days a so-called "dead" bank was carried and during the last four days a floating bank. During the day period the boiler was operated at approximately 157 per cent of rating. A dead bank as used here is one in which the coal feed is entirely shut off and the fire is allowed to burn back as far as possible without permitting it to go out entirely. Small amounts of coal, just sufficient to keep the fire, are fed from time to time as required. During a dead bank a boiler will cease to steam, and the pressure will fall to atmospheric if the banking period is of sufficient duration. With a floating bank the coal feed is sufficient to keep the tuyeres well covered and to hold a fire which will maintain full pressure in the boiler. With a floating bank the boiler may even continue to steam at a very low rating throughout the banking period. Each banking test was divided up into overlapping periods

with a different proportion between steaming and banking hours. The coal required for banking was computed by taking the difference between the coal actually used during a period and the quantity computed as being required to evaporate the water fed to the boiler during the same period. An average heating value of the

TABLE 3 COMPARISON OF COAL ANALYSES

(Analyses are for dry coal and are made on duplicate samples. Samples Nos. 106, 116 and 126 are Shipper "A" coal; sample No. 138, Shipper "B" coal)

Test No.	Sample No.	Analysis made by	Proximate Analysis				Ultimate Analysis					
			Fixed carbon	Volatile matter	Ash	B.t.u.	C	H_2	O_2	N_2	S_2	Ash
4	106	D	55.30	33.50	11.20	13133[1]	73.55	4.44	8.76	1.30	0.75	11.20
		S	54.14	34.93	10.93	13297	70.57	5.15	10.93	1.38	1.04	10.93
		P	54.96	34.40	10.64	13062	73.67	4.72	8.44	1.37	1.16	10.64
9	116	D	55.15	34.55	10.30	13365[1]	74.59	4.82	7.85	1.54	0.90	10.30
		S	53.88	36.08	10.04	13284	74.91	5.13	7.83	1.14	0.98	10.04
		P	54.38	35.34	10.28	13150	74.10	4.98	8.09	1.44	1.11	10.28
14	126	D	54.90	34.85	10.25	13352[1]	75.12	4.72	7.45	1.32	1.14	10.25
		S	53.40	35.87	10.73	13298	71.94	5.30	9.26	1.44	1.33	10.73
		P	55.28	34.03	10.69	13146	72.48	4.98	9.10	1.43	1.32	10.69
24	138	D	58.75	33.50	7.75	13927[1]	75.83	5.08	9.06	1.25	1.03	7.75
		S	55.54	35.65	8.81	13520	75.88	5.90	7.01	1.26	1.14	8.81
		P	57.11	33.93	8.96	13584	77.63	5.19	5.56	1.41	1.25	8.96

[1] Values used in computations. D, The Detroit Edison Co.; S, Solvay Process Co.; P, Pittsburgh Testing Laboratory.

coal used during each method of banking was taken as the standard, and all coal quantities were corrected to this standard heating value.

22 The last three tests conducted (Nos. 101-103) were strictly operating tests to determine the results of different methods of operating the boilers during the low-load period. Throughout these tests the steaming rate was varied according to the plant load, the boiler carrying its share of the total load during the day period. Test No. 101 was of 48 hr. duration with the boiler steaming at a low rate during the night. Test 102 was also of 48 hr. duration, but the boiler was subjected to a dead bank at night. Test 103 was of 25 hr. duration and the boiler was steamed at a rating of 90 per cent during the night.

23 Throughout the tests the data were copied from the original log sheets on to three large special data sheets each hour so that any variation from uniform conditions could be traced and the necessary changes in operation made. A graphical log was maintained for the more important items. All data tabulated [1] have been corrected for instrumental errors and in addition the coal and water weights have been corrected for the amount of coal taken for sampling and the water leakage from pump glands, safety valves, and water columns. A correction was also made for the difference in level of water in the feedwater receiving tank and in the boiler at the beginning and end of each test, and for blowdown. It was not always possible to dump ashes at the beginning and end of each test, so where the weights were for longer periods they were prorated on a coal-weight basis. Ash was usually weighed dry, but when wetting was found necessary a moisture sample was taken and the weight was corrected to a dry basis.

COAL, REFUSE AND ASH SAMPLING

24 Car samples were taken of the coal intended for the test and ash determinations were made so that those falling outside of prescribed limits could be diverted to other uses. In this way it was possible to control the quality of coal used during each series of tests. Coal samples were taken every 30 min. during the test at the stoker hoppers, and at the weighing scales each time a bucket of coal was dumped. Each sample was of such an amount as to approximate one-tenth of one per cent of the coal burned. Due to the segregation of coal coming down the coal spreaders to the stoker, samples were taken at twelve places along the hoppers in order to assure a representative sample. The method of taking samples at the scales was such that the sample contained a slightly larger percentage of fines than the average coal, with the result that the analysis gave a higher ash and lower heating value than was obtained from analysis of the hopper sample. For this reason the analyses obtained from the hopper samples were used in all computations except in a few instances where the scale sample analyzed higher in B.t.u.; then an average was taken between this sample and the reserve hopper sample, which was always prepared for

[1] Extended tables containing further data and results obtained in these tests will be found in the discussion following this paper.

analysis but not analyzed except for the purpose mentioned or for checking purposes. Separate moisture samples were taken hourly in duplicate and immediately sealed. This duplicate sample was taken for the same purpose as the duplicate hopper sample: to be used in case of the loss of a sample or for checking purposes.

25 The refuse sample was taken each time the ashes were dumped, the amount being approximately two-tenths of one per cent of the total weight of refuse.

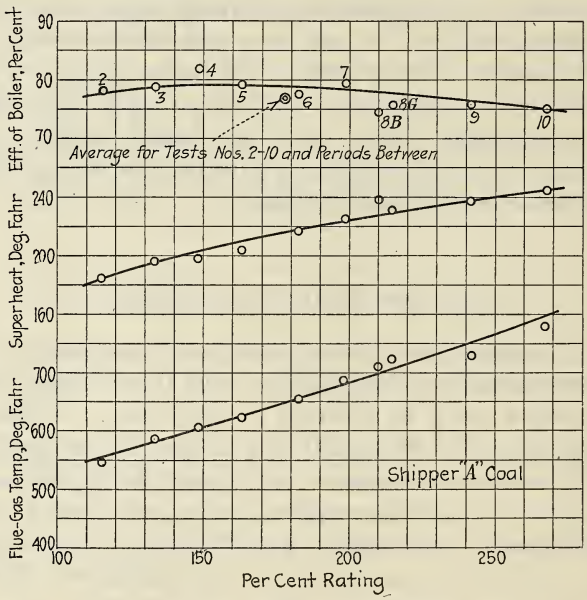

Fig. 7 Results Obtained with Original Baffle

26 All laboratory analyses were made in the chemical laboratory of The Detroit Edison Company, where all plant samples are analyzed regularly. In order to check the results obtained, four samples were sent to the laboratory of the Solvay Process Company and to the Pittsburgh Testing Laboratory for analysis. These were samples 106, 116, 126 and 138, taken during tests Nos. 4, 9, 14 and 24. The results of the analyses as reported are given in Table 3. In order to check the method of coal sampling employed in the test, an investigation was conducted by the Research Department of The Detroit Edison Company. The results of this investigation

proved that the methods employed were accurate and that a representative sample was obtained for analysis.

27 Except during the special flue-blowing tests, flues were blown every eight hours. The front bank of tubes was blown by hand lance when necessary, or about once each week. Soot was removed from the soot chamber at the bottom of the last pass every night except in a few instances where the last test before burning out the fire was to end the following morning.

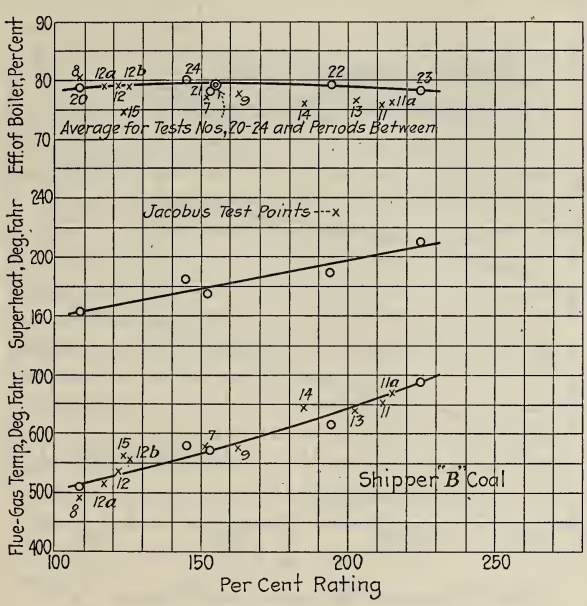

Fig. 8 Results Obtained with Original Baffle

28 Before starting a test the fire was brought to its proper condition for the particular rating to be carried, and when it was thought that conditions had become practically constant the test was started. Throughout the duration of the test the coal was maintained at its proper thickness on the tuyeres, and at the completion of the test it was in all cases, as nearly as could be judged, the same as at the start. During the periods between tests and during periods of starting up and burning out fires, all important observations were taken. This made possible the combining of several tests so that an overall efficiency for each series of tests and

parts of series of tests could be computed.[1] The duration of each test was determined from the total weight of coal burned. In estimating the thickness of a fire containing as much as 15 tons of coal it is conceivably impossible to detect a difference of less than $1\frac{1}{2}$ tons. Since it was desired that the errors in judgment of the thickness of fuel bed at the beginning and end of a test should not introduce errors greater than 1 per cent, the total weight of coal burned necessarily must be 300,000 lb. or more. Some tests, how-

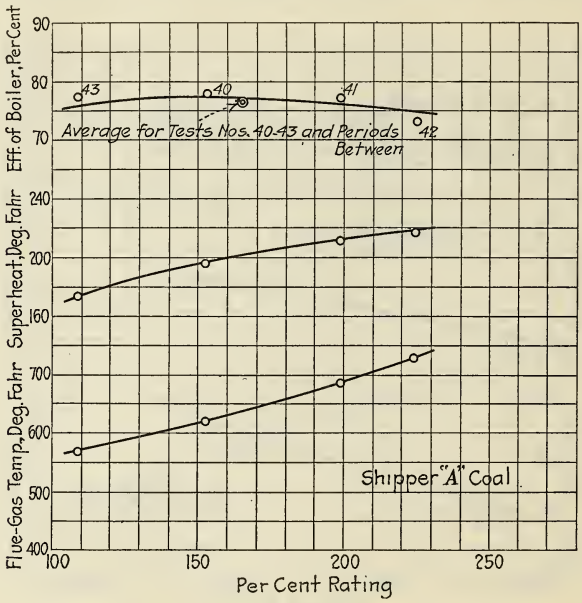

Fig. 9 Results Obtained with Baffle "A"

ever, were terminated before this quantity of coal was burned, due either to the fact that conditions appeared to be so constant as to warrant the shorter period, to a shortage of the special coal being used, to hot grinders, or to an overheated setting.

29 Marks and Davis' steam tables were used in computations. All computations involving the heating value of the coal are on an "as-fired" basis.

DISCUSSION OF RESULTS

30 *Baffle Arrangements.* Figs. 7, 8, 9, 10 and 11 show the results of all the constant-rating tests in graphical form for the

[1] See footnote on page 1017.

different baffle arrangements. Fig. 7 shows that before any changes had been made to the baffling the efficiency touched 79 per cent at 160 per cent rating. After the boiler was washed the efficiency curve was raised about 0.75 per cent, with a reduction in superheat. With the original baffling, Shipper "B" coal gave the highest efficiencies. These are shown in Fig. 8, in which are also plotted the points as reported by Dr. Jacobus in 1911 for the purpose of comparison. Fig. 10 shows the results obtained with the boiler baffled according to the "B" arrangement. These efficiencies were

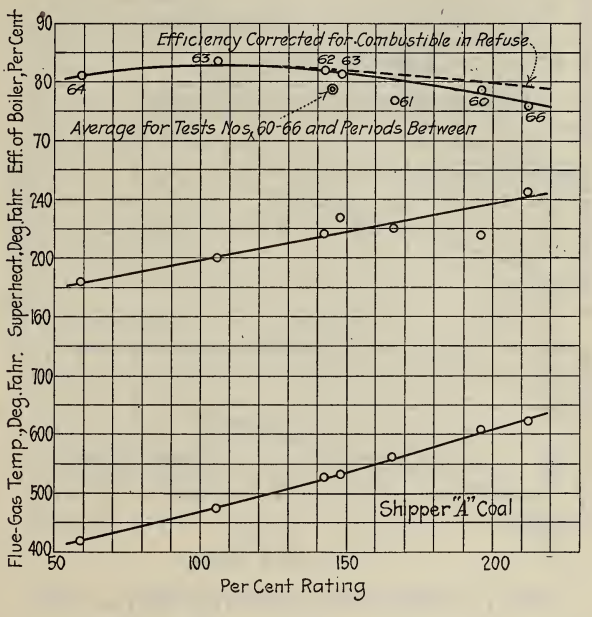

Fig. 10 Results Obtained with Baffle "B"

higher than with the original baffling, touching 83.5 per cent at 106.2 per cent rating. Due to the fact that the extension grates, which will be mentioned later, had been changed prior to this series of tests for others with larger openings, the combustible in the refuse was greater during the high-rating tests than with the original baffle. In order to correct for this loss, which is not chargeable to the baffle, a broken line has been drawn in to show what the efficiency would have been with an ashpit loss the same as during the first series of tests. With the "C" baffle arrangement the

maximum efficiency reached was 82.4 per cent at 130 per cent rating. The results of this series of tests are shown in Fig. 11.

31 Each time the baffles were changed the row of tubes nearest the combustion chamber was partially cleaned by the workmen in making the changes, whereas the other tubes which were not touched would retain their soot deposit. When comparing "B" and "C" with the original baffling this difference in condition of heating surface should be kept in mind. The "B" baffle shows a lower

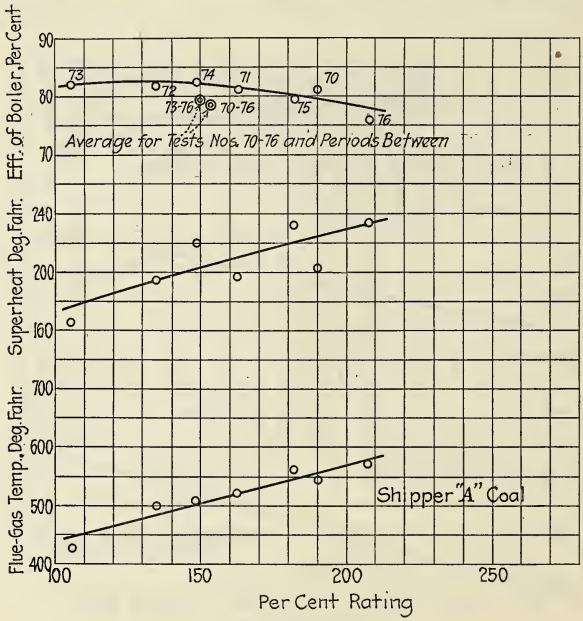

Fig. 11 Results Obtained with Baffle "C"

stack-gas temperature and higher superheat which at its best point gives an efficiency $2\frac{1}{2}$ per cent above the efficiency obtained with the original baffling; the best point, however, being at a lower rating than before. Comparing the average efficiencies obtained during the entire series of tests on the original baffle before washing with the average obtained during the entire series of tests on the "B" baffle arrangement, we find that the original baffle gave 76.8 per cent at 178 per cent rating and the "B" baffle 79.1 per cent at 145 per cent rating. These results include all periods between the tests of the respective series, and one starting and burning-out

period. With the original baffling after washing, the first five tests (Nos. 11–15) gave an average efficiency of 79.5 per cent at 99 per cent rating, and tests Nos. 16–18 gave 79.7 per cent at 122.8 per cent rating. From the results as shown in the heat balance (Table 6), it is apparent that these efficiencies are too high.

32 A comparison between the average results obtained with the "B" and "C" arrangements shows the former to be 79.5 per cent at 145 per cent rating and the latter 79.8 per cent at 157 per

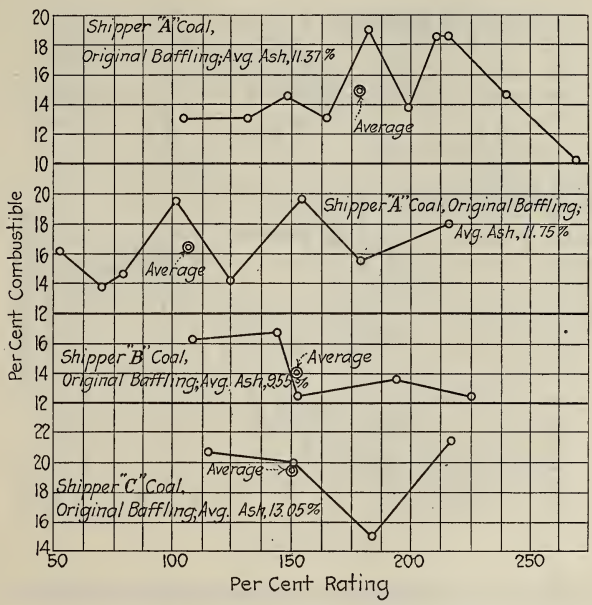

Fig. 12 Percentage of Combustible in Refuse

cent rating. The gain, however, is not believed to be due entirely to the baffle arrangement, as a separation of the furnace, grate, and boiler losses show that for the "C" arrangement the unavoidable losses were 0.7 per cent lower than for the "B." Granting that a gain of 0.3 per cent in efficiency is accounted for by the "C" baffle arrangement, the fact that the "B" baffle gave a higher superheat makes it more desirable from the standpoint of overall plant economy, due to the effect of additional superheat on reducing the turbine water rate.

33 *Extension Grates.* During the tests, extension grates with different-sized openings were installed with the object of reducing

the combustible in the refuse by furnishing more air over the grinders. Three different arrangements were made, including the standard grates which were in use at the beginning of the test. These original grates had a total area of openings of 11.76 sq. in. per set. At the completion of test No. 10 new grates having a total area of openings of 17.15 sq. in. were installed. Just before test No. 40 another change was made by increasing the air openings through the five top bars, leaving the two bottom bars as before. The total area of

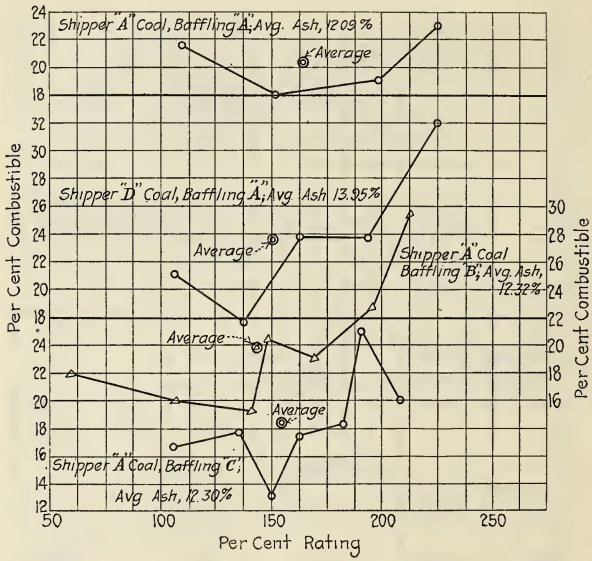

FIG. 13 PERCENTAGE OF COMBUSTIBLE IN REFUSE

openings per set was then 25.40 sq. in. The damper control was also changed and improved to give closer regulation of the air to the extension grates. This last change did not give the expected results since the combustible in the refuse increased from 3 to 5 per cent over that obtained with the original grates. Although the ash content of the coal was about 1 per cent greater than during the first series of tests, it is not believed that this could account for the increase in combustible in the refuse, since the average rating carried with the original grates was considerably higher than after the second change. Tests Nos. 40–43 show the greatest loss of combustible, which is believed to be due to the fact that these being the first tests after the change, the operators were not entirely familiar

with the method of operation. Also during two of these tests the fusion temperature of the ash was rather low, resulting in some trouble with large clinkers at the high ratings. Due to these facts it was considered advisable not to use the results obtained during tests Nos. 40 to 43 in the comparison. The effect of the second change in extension grates should be determined from tests Nos. 60–67 and 70–76. Comparing the combustible for those tests (Fig. 13) with that obtained before and after the first change (Fig.

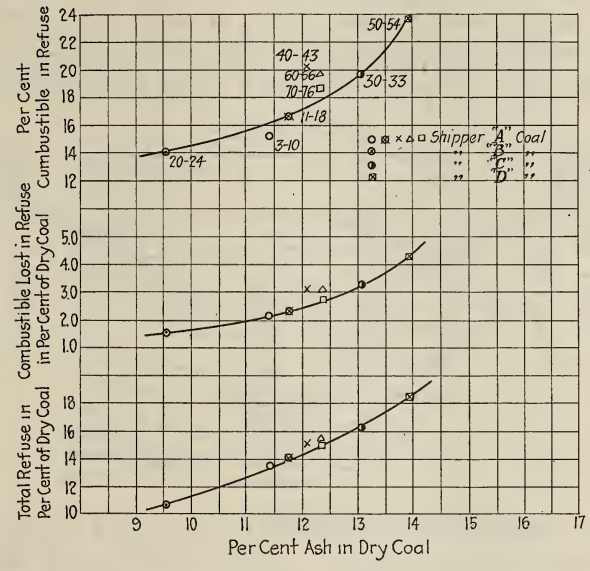

Fig. 14 Relation Between Ash Contained in Coal and Combustible Lost in Refuse

12), it is seen that at low ratings the percentage after the second change was only very little greater than after the first one, although from 2 to 3 per cent greater than with the old type, and that the high averages are caused by greatly increased losses at higher ratings. That this increased loss due to combustible in refuse was due to the arrangement of air openings in the extension grates, is verified by the fact that at the completion of all tests the boiler was put back into regular plant operation, and the same difficulty with down blast through the grinders was experienced as during the test. Grates of the old type were then installed and additional observations were made at the same ratings as before, with the result that

there was no down blast and the refuse contained about the same percentage of combustible as during tests Nos. 2–10. Figs. 12, 13, and 14 show the percentage of combustible for the different tests and also the average percentage for each series of tests. A sketch of the stoker and fuel bed is shown in Fig. 16.

34 *Air Supplied to Stoker.* It will be seen from the results obtained from the Thomas air meter that this method of measuring the air supplied for combustion was not accurate, due to the leakage of air from the air chamber beneath the stoker. The results indi-

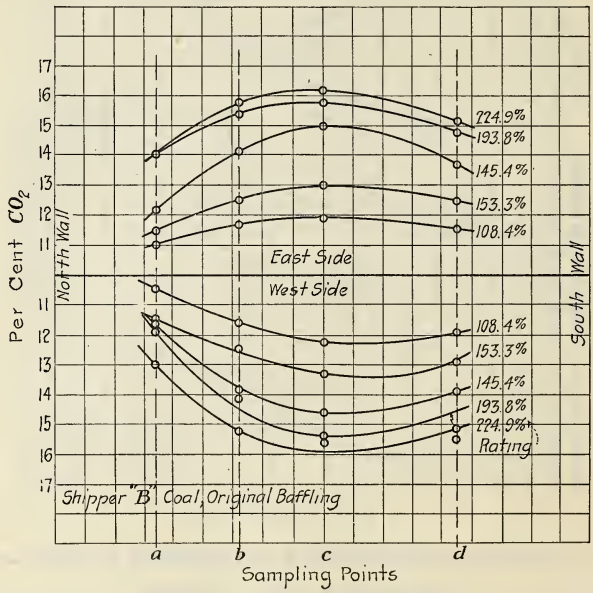

Fig. 15 Variation of CO_2 Across Boiler at Damper

cate that at high ratings the blower supplied over 25 per cent more air than was actually delivered through the stoker tuyeres. The Thomas air meter was checked against a 36-in. orifice with rounded entrance and was found to be an accurate means of measuring the quantities of air supplied by the forced-draft fan, but due to leakage from the air chamber the results could not be used for combustion computations.

35 *Flue-Gas Analysis.* Fig. 15 shows how the CO_2 varies at different points across the path of flow of the gas at the damper. The vertical broken lines represent the locations of the sampling

tubes. The points plotted are the averages for the tests made using Shipper "B" coal and with the original baffling.

36 *Results Obtained with Different Coals.* Data on the four different coals used in the tests are given in Table 2. Inasmuch as the different coals were not tested under identical conditions as regards boiler and stoker, the efficiencies obtained are not directly comparable. Except for Shipper "D" the percentage of CO_2 was about the same for the different coals, so the comparison may be made on the basis of the ashpit loss only. In Figs. 12 and 13 the percentage of combustible in the refuse is plotted against boiler rating. Values from the individual tests vary, due partly to the difficulty in obtaining representative refuse samples and partly to the effect pro-

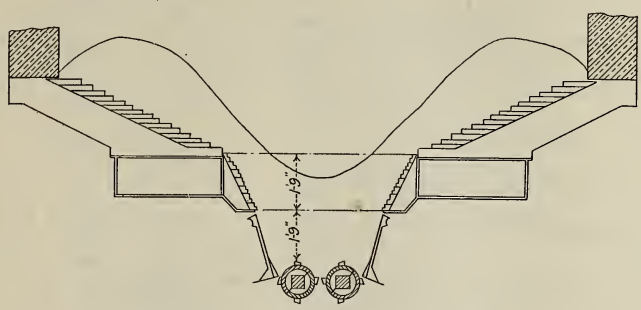

Fig. 16 Shape of Fuel Bed

duced by the blowing of soot from the hopper under the back pass of the boiler into the ash hopper. The combustible content of the soot was greater than that of refuse, so that if the soot happened to be blown an unequal number of times during different tests, the combustible content of the refuse was influenced by the presence of different quantities of soot. It is believed, however, that by making a comparison on the basis of the average percentages of combustible for each series of tests, these variations should be less apparent. The averages for all tests and for periods between tests are shown in Figs. 12 and 13, and in Fig. 14 the same averages have been plotted against the percentage of ash in the coal. Except for the last series of tests (Nos. 70 to 76) with Shipper "A" coal, the points fall fairly well on a smooth curve. As explained before under the discussion of extension grates, it is believed that this increased loss was caused by the arrangement of air openings in the extension grates. The actual heat lost is shown in Fig. 17 and varies from 230 to 530 B.t.u.

per lb. of coal. In this same figure are shown the loss in per cent of heating value of the coal, the B.t.u. per lb. of coal, the B. t.u. available for the boiler, and the relative value of coal on a B.t.u. basis in per cent of a coal with an arbitrary standard of 10 per cent ash.

37 *Pass Temperatures.* Figs. 1, 2, 3, and 4 show the points on a cross-sectional view of the boiler at which temperatures were taken, and Table 4 gives the temperatures at the different points. The readings of the thermocouples at symmetrical points in the two

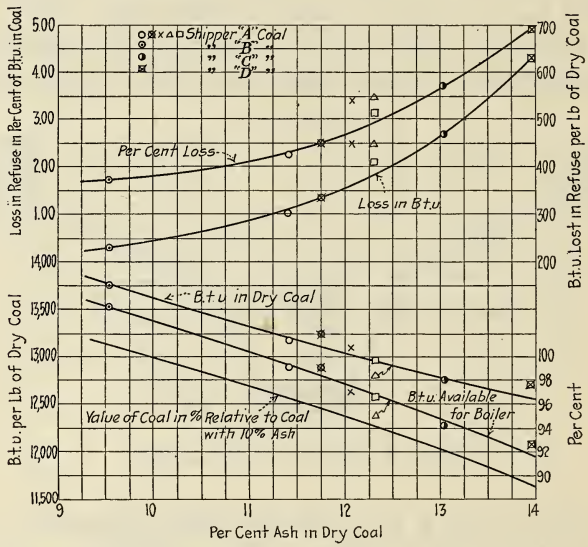

FIG. 17 HEAT LOSS DUE TO COMBUSTIBLE IN REFUSE

halves of the boiler have been averaged, and these averages are given in the table instead of the temperatures at each individual point. The over-fire temperature (point *F*) is the average of the two readings taken, one at the north and one at the south end of the combustion chamber. Points 15–16 are the average of the two readings 15 and 16, while every other point is an average of four; for instance, a point marked 3–4 is the average of thermocouples 3*S*, 3*C*, 4*N* and 4*C*. Observations obtained at point *F* are not correctly the temperature in the furnace due to the radiant heat given to the front rows of boiler tubes.

38 With the original baffling, thermocouples at points 9 and 10 were not in the direct path of the gas and therefore did not indi-

TABLE 4 PASS TEMPERATURES

Test No.	% Rating	Pass Temperatures, Deg. Fahr.									Temperature at damper, deg. fahr.	Over-fire temperature, deg. fahr.
		Thermocouple Nos.										
		1&2	3&4	5&6	7&8	9&10	11&12	13&14	15&16	17&18		
2	116.0	1206	968	932	.....	581	547	497	.....	.....	547	1623
3	133.0	1274	1019	980	.....	628	576	513	.....	.....	587	1676
4	148.5	1320	1053	1002	.....	728	589	526	.....	.....	601	1708
5	163.7	1337	1071	1046	.....	671	605	547	.....	.....	623	1739
6	182.0	1398	1142	1109	.....	722	641	576	.....	.....	651	1756
7	199.0	1437	1190	1152	.....	760	666	597	.....	.....	685	1785
8A	215.0	1503	1238	1203	.....	819	705	630	.....	.....	719	1850
8B	210.0	1482	1222	1202	.....	830	710	621	.....	.....	710	1819
9	241.0	1588	1287	1257	.....	860	743	651	.....	.....	733	1899
10	268.0	1626	1335	1304	.....	914	782	651	.....	.....	782	2060
11B	79.0	799	721	730	.....	495	476	434	436	.....	473	.....
12	102.8	901	773	791	.....	533	494	462	469	.....	494	1260
13	53.2	685	655	664	.....	474	447	432	418	.....	447	.....
14	126.2	1029	848	846	.....	566	509	495	486	.....	504	1435
15	154.5	1149	917	928	.....	612	547	533	528	.....	546	1574
16	180.0	1268	1041	1026	.....	680	602	591	575	.....	607	1620
17	67.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	469	.....
18	216.7	1373	1102	1094	.....	722	638	627	608	.....	645	1740
20	108.4	1024	838	852	.....	561	509	496	487	.....	510	1383
21	153.3	1159	927	948	.....	623	562	549	543	.....	572	1534
22	193.8	1299	1034	1046	.....	685	612	599	591	.....	611	1652
23	224.9	1421	1146	1152	.....	773	685	681	657	.....	690	1778
24	145.4	1216	989	968	.....	637	565	557	543	.....	580	1540
30	151.4	1236	1001	977	.....	648	571	560	547	.....	560	1596
31	115.0	1118	916	890	.....	578	521	507	496	.....	530	.....
32	184.1	1284	1078	1055	.....	695	601	591	572	.....	619	1771
33	216.5	1428	1137	1126	.....	754	647	637	616	.....	655	1835
40	153.5	1190	964	972	.....	736	633	580	596	.....	614	1612
41	199.0	1320	1056	1076	.....	827	711	659	670	.....	691	1778
42	225.8	1421	1127	1154	.....	905	771	716	722	.....	726	1835
43	108.0	1034	876	868	.....	686	590	537	555	.....	566	.....
50	106.1	1077	890	888	.....	696	629	545	564	.....	571	.....
51	137.0	1196	976	951	.....	738	631	575	587	.....	604	1515
52	163.6	1296	1043	1029	.....	799	687	634	641	.....	660	1647
53	194.3	1374	1118	1089	.....	846	721	669	671	.....	682	1648
54	226.4	1427	1185	1173	.....	934	793	751	747	.....	751	1698
60	196.4	1338	1053	1037	746	734	648	567	623	1157	601	1703
61	168.4	1299	1022	990	712	678	607	538	580	1114	558	1671
62	142.0	1272	1050	953	676	629	568	499	538	1080	526	1669
63	106.2	1159	920	844	600	551	504	443	479	978	466	1561
64	59.0	953	781	705	535	484	455	407	428	824	407	.....
65	148.5	1284	1015	962	691	635	567	503	542	1092	527	1733
66	212.5	1413	1118	1102	826	760	679	591	638	1227	620	1769
70	190.0	1339	1185	1012	680	620	663	524	571	677	547	.....
71	162.0	1282	1119	949	629	576	642	487	548	641	522	.....
72	134.0	1239	1086	914	607	557	627	475	528	634	499	.....
73	105.3	1111	966	821	543	516	519	423	468	588	424	.....
74	149.0	1293	1128	944	636	619	603	497	539	667	507	.....
75	183.2	1393	1256	1087	698	679	653	536	577	725	558	.....
76	208.5	1443	1298	1129	731	707	682	552	589	754	566	.....

cate the correct temperature. After the first baffle change the couples were more nearly in the center of the pass. In Fig. 18, which shows the variation in temperatures through the passes plotted against the percentage of heating surface passed over, points 9 and 10 for the original baffling appear to be too low. The same applies to points 3 and 4, except that the baffle change did not correct the error and it was found advisable to install additional thermocouples 17 and 18, as shown in Fig. 3. Owing to the proximity

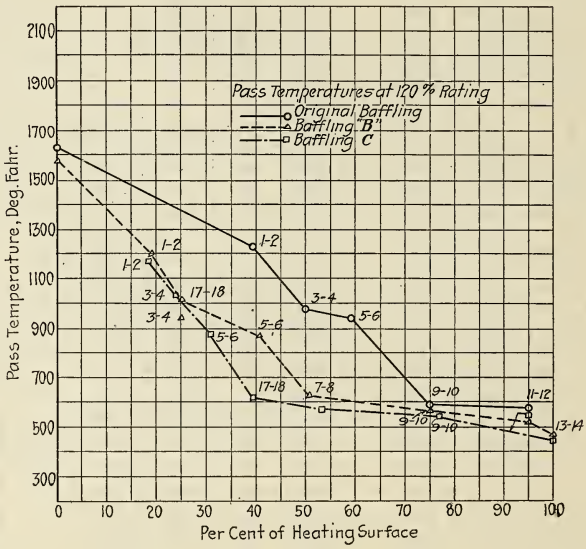

Fig. 18 Pass Temperatures

of thermocouples 11–12 and 13–14, they should have given approximately the same temperature, whereas the thermocouples at points 11 and 12 indicated a higher temperature than those at 13 and 14. This is accounted for by the fact that there is a very narrow space between the tubes at the point where they enter the upper outside drums and the baffle cannot be made absolutely tight, which permitted hot gas from the superheater pass to leak through and pass over thermocouples at points 11 and 12. Several attempts were made to move thermocouples 13 and 14 to find the best location, but 11 and 12 could not be moved, as they were entered through the end walls of the setting. The heating surface, as shown in Fig. 18 and in Table 5, was computed on the basis of effective heating sur-

face, assuming the superheater surface to have the same effectiveness in absorbing heat as the remainder of the boiler. Computations were made which showed the superheater absorbs about the same amount of heat per square foot as the average for the boiler.

39 *Heat Losses.* Table 6 gives the losses itemized in percentages of the total heat supplied in the coal. The loss due to combustible in refuse is based upon the percentage of combustible in the refuse and is computed by assuming that the refuse lost through the stack contains the same percentage of combustible as the refuse from

TABLE 5 COMPUTED PERCENTAGE OF HEATING SURFACE PASSED BY GASES

Thermocouples	Percentage of heating surface passed over by flue gases			
	Original baffle	"A" baffle	"B" baffle	"C" baffle
Over-fire	0	0	0	0
Nos. 1 & 2	38.8	38.8	18.8	18.2
Nos. 3 & 4	50.6	50.6	28.2	23.4
Nos. 17 & 18	.....	.....	28.2	38.8
Nos. 5 & 6	58.5	58.5	41.2	31.8
Nos. 7 & 8	.....	.....	51.2	53.3
Nos. 9 & 10	74.6	74.6	71.2	76.6
Nos. 11 & 12	95.0	95.0	95.0	95.0
Nos. 13 & 14	100.0	100.0	100.0	100.0
Nos. 15 & 16	100.0	100.0	100.0	100.0

the ashpit. Presumably the refuse to the stack contains more combustible than was shown by the analysis of the ashpit refuse, so the losses under this heading probably are slightly under the correct values.

40 In the radiation and unaccounted-for losses are included radiation from the boiler setting, loss due to sensible heat in refuse, loss due to unburned gases other than CO, and all errors involved in observation and analysis. Due to the fact that this item in most cases is small compared with results obtained in other tests, a few words of explanation are given to show that the radiation on this type of boiler is very low. Mr. H. Kreisinger [1] gives the radiation per square foot of exposed surface as 250 B.t.u. per hr. at 100 per cent rating and 350 B.t.u. per hr. at 200 per cent rating. The

[1] Boiler Tests with Pulverized Illinois Coal, H. Kreisinger and J. Blizard, *Mechanical Engineering*, May, 1921, p. 321.

total exposed surface of this boiler, including steam drums, is 4294 sq. ft., or 1.81 sq. ft. per rated boiler hp., giving a radiation loss per hour per rated boiler hp. of 453 B.t.u. at 100 per cent rating and 634 B.t.u. at 200 per cent rating. The coal burned per rated boiler hp-hr. is practically a straight-line function between 50 per cent and 225 per cent rating, which gives 3.4 lb. per hr. at 100 per cent rating and 6.77 lb. per hr. at 200 per cent rating. This gives the

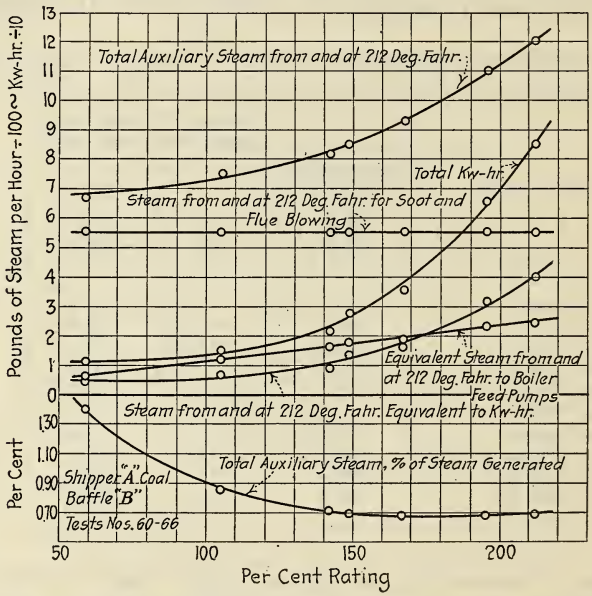

Fig. 19 Auxiliary Energy Required per Hour

radiation loss per pound of coal as 133 B.t.u. (1.06 per cent) at 100 per cent rating and 95 B.t.u. (0.76 per cent) at 200 per cent rating, assuming a coal of 12,500 B.t.u. per lb. as fired. An effort was made to check these values by computing the radiation losses on the bases of the exposed surfaces and certain temperatures, some of which were actually measured on a boiler of this type, and some assumed. Conservative values of heat-transmission coefficients were taken, with the result that the calculation gave the radiation per pound of coal as 75 B.t.u. (0.60 per cent) at 100 per cent rating and 40 B.t.u. (0.32 per cent) at 200 per cent rating. With the exception of a few tests which were of two short duration or were otherwise unsatisfactory, the observed radiation losses did not

TABLE 6 HEAT BALANCE

Test No.	% Rating	Heat absorbed by boiler	Loss due to moisture in coal	Loss due to hydrogen	Heat Lost to Stack in				Loss due to incomplete combustion	Loss due to combustible in refuse	Radiation and unaccounted for	B.t.u. in coal as fired
					Theoretical dry gas	Excess dry air	Moisture in air	Total				
2	116.0	77.9	0.3	4.5	8.9	3.0	0.2	12.1	0.5	1.9	2.8	12690
3	133.0	78.3	0.4	4.8	9.5	3.7	0.1	13.3	0.6	1.9	0.7	12650
4	148.5	81.2	0.5	3.9	9.7	3.9	0.2	13.8	0.6	2.0	−2.0	12425
5	163.7	78.9	0.5	4.5	10.1	3.2	0.2	13.5	0.4	1.8	0.4	12750
6	182.0	77.3	5.6	4.3	10.9	3.4	0.1	14.4	0.4	3.0	−0.03	12550
7	199.0	79.8	0.6	4.5	11.7	2.7	0.1	14.5	0.7	2.3	−2.4	12277
8A	215.0	75.4	0.7	4.4	12.0	3.0	0.2	15.2	0.6	3.0	0.7	12432
8B	210.0	74.4	0.5	4.4	11.8	4.1	0.2	16.1	0.7	3.0	0.9	12595
9	241.0	75.4	0.7	4.4	12.1	2.6	0.2	14.9	0.5	1.9	2.2	12550
10	268.0	75.2	0.6	4.6	13.0	2.9	0.3	16.2	1.1	1.2	1.1	12540
11B	79.0	75.8	0.5	4.1	7.2	5.1	0.3	12.6	0.5	1.8	4.7	12780
12	102.8	78.1	0.6	4.3	7.1	5.0	0.2	12.2	0.5	3.1	1.1	12563
13	53.2	76.0	0.4	4.3	7.0	6.6	0.2	13.8	0.6	2.3	2.6	12860
14	126.2	83.1	0.6	4.0	8.2	2.1	0.1	10.4	0.4	1.8	−0.3	12580
15	154.5	80.3	0.4	4.2	8.9	2.6	0.2	11.6	0.4	3.8	−0.8	12677
16	180.0	80.5	0.4	4.0	9.8	3.1	0.2	13.1	0.2	2.6	−0.8	12425
17	67.4	79.5	0.5	3.9	7.2	5.0	...	12.2	0.5	2.1	1.3	12730
18	216.7	80.5	0.3	4.0	10.6	2.8	0.2	13.6	0.3	2.9	−1.6	12690
20	108.4	79.0	0.4	4.1	8.1	4.2	0.2	12.5	0.5	2.0	1.7	13100
21	153.3	78.3	0.4	3.6	9.1	4.0	0.1	13.2	0.4	1.4	2.7	13280
22	193.8	79.9	0.4	4.7	9.6	1.8	0.1	11.5	0.5	1.7	1.2	13220
23	224.9	78.1	0.3	4.3	11.2	1.9	0.2	13.3	0.6	1.5	1.8	13330
24	145.4	80.0	0.3	4.2	7.7	4.3	0.2	12.2	0.4	1.6	1.3	13540
30	151.4	78.8	0.4	4.0	9.1	2.9	0.2	12.2	0.4	3.5	0.7	12305
31	115.0	78.6	0.3	4.2	8.4	2.6	0.2	11.2	0.2	4.0	1.5	12283
32	184.1	75.6	0.4	4.1	9.8	2.4	0.3	12.5	0.3	2.6	4.5	12300
33	216.5	77.1	0.3	3.8	10.8	2.7	0.3	13.8	0.3	4.1	0.6	12390
40	153.5	77.9	0.5	4.3	10.2	3.9	0.2	14.3	0.3	3.3	−0.6	12240
41	199.0	76.7	0.5	4.5	11.5	3.7	0.3	15.5	0.2	3.1	−0.5	12580
42	225.8	72.9	0.4	3.9	12.5	3.4	0.3	16.2	0.4	3.6	2.6	12813
43	108.0	77.3	0.5	4.2	9.3	6.2	0.2	15.7	0.5	4.1	−2.3	12357
50	106.1	77.5	0.5	4.2	9.2	6.6	0.2	16.0	0.3	4.8	−3.3	11913
51	137.0	76.0	0.5	4.3	9.8	4.4	0.2	14.4	0.2	3.6	1.0	12072
52	163.6	75.0	0.4	3.7	10.9	4.3	0.3	15.5	0.2	4.6	0.6	12380
53	194.3	76.0	0.3	4.3	10.9	2.6	0.3	13.8	0.3	4.7	0.6	12530
54	226.4	70.9	0.4	4.1	12.4	2.5	0.4	15.3	0.5	7.3	1.5	12260
60	196.4	78.5	0.5	4.1	9.8	1.6	0.3	11.7	0.4	4.3	0.5	12200
61	168.4	77.0	0.4	4.1	8.8	1.7	0.4	10.9	0.4	3.1	4.1	12445
62	142.0	81.7	0.5	4.5	8.3	1.4	0.2	9.9	0.4	2.5	0.5	12240
63	106.2	83.5	0.5	3.9	7.0	1.7	0.2	8.9	0.4	2.7	0.1	12261
64	59.0	80.9	0.3	4.2	6.1	3.3	0.2	9.6	0.1	2.8	2.1	12549
65	148.5	80.8	0.4	4.7	8.0	1.1	0.3	9.4	0.4	3.8	0.5	12220
66	212.5	75.8	0.4	3.8	10.3	1.5	0.3	12.1	0.4	5.6	1.9	12290
70	190.0	80.4	0.3	3.7	8.5	2.0	...	10.5	0.4	4.4	0.3	12730
71	162.0	81.3	0.4	3.6	7.8	2.2	0.4	10.4	0.4	3.3	0.6	12340
72	134.0	82.0	0.5	3.7	7.4	1.9	0.4	9.7	0.4	3.3	0.4	12149
23	105.3	82.2	0.3	3.6	6.3	1.8	0.2	8.3	0.3	2.6	2.7	12550
74	149.0	82.4	0.6	3.8	7.7	2.3	0.2	10.2	0.3	2.1	0.6	12118
75	183.2	79.8	0.6	3.7	8.8	2.2	0.2	11.2	0.3	2.7	1.7	12410
76	208.5	75.6	0.5	3.6	8.6	2.0	0.3	10.9	0.4	3.2	5.8	12650

depart from the computed radiation by more than 1.5 per cent of the heating value of the coal.

ENERGY USED BY BOILER AUXILIARIES

41 The stokers, blowers, and grinders were driven by direct-current variable-speed motors, the electrical input to which was measured by watthour meters. This electrical energy was all gen-

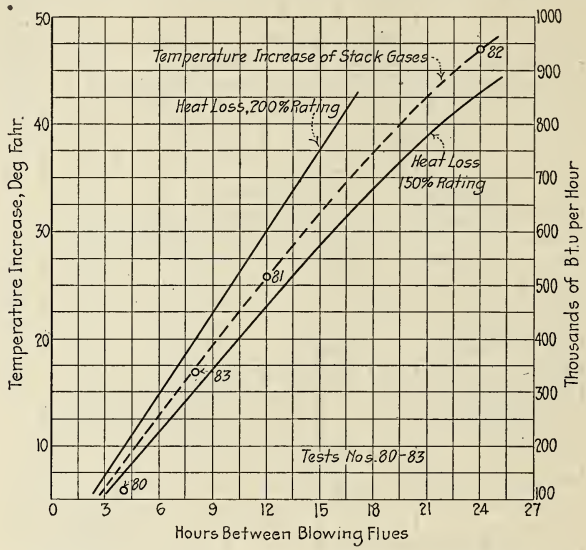

FIG. 20 EFFECT OF VARYING TIME INTERVAL BETWEEN BLOWING OF FLUES

erated on the auxiliary turbo-generators, the exhaust steam from which is all utilized for heating feedwater. Tests have been made which indicate a cost of 4550 B.t.u. per kilowatt-hour generated on these auxiliary turbo-generators. The steam equivalent to the electrical energy has therefore been computed on this basis and reduced to equivalent steam from and at 212 deg. fahr.

42 The steam required to drive the 1200-gal.-per-min. boiler-feed pump was metered; but the pump was much larger than was needed to serve one boiler, which necessitated operating the pump at low ratings with a correspondingly high steam consumption. From tests it has been shown that when the exhaust from normally loaded boiler-feed pump turbines was utilized for heating the feedwater, the cost of pumping water was 1.7 B.t.u. per lb. of water. Expressed in equivalent steam from and at 212 deg. fahr., the pounds

of steam per hour is then equal to 0.00175 multiplied by the pounds of water pumped per hour.

43 Since no relation appears to exist between the quantity of steam required for flue and soot blowing and the percentage rating, these quantities are taken as constant hourly quantities over the testing periods. By taking an average of the quantities of flue-blowing steam which were metered and computing the quantities required for soot removal, a figure of 555 lb. per hr. equivalent steam from and at 212 deg. fahr. was obtained as the amount required for both flue blowing and soot removal.

44 Fig. 19 gives for tests Nos. 60–66 the electrical energy required per hour for the blowers, stokers, and grinders, the steam equivalent to the electrical energy, and the total steam for all auxiliary uses expressed as equivalent steam from and at 212 deg. fahr.

FLUE BLOWING

45 Four tests (Nos. 80–83) were made to determine what effect varying the time interval between blowing flues would have on the temperature of flue gas leaving the boiler. The test data and results are given in Table 7. It was found necessary to plot the hourly readings of CO_2, temperature of gas at damper, and the steam flowmeter readings for each test and then draw in an average temperature curve for the period following the blowing of flues in order to obtain the temperature difference between the beginning and end of the period.

46 By considering only periods during which conditions were uniform, and arranging these for each test, the following values were obtained:

Test No.	Time interval between blows	Temperature increase, deg. fahr.
80	4	6
81	12	26
82	24	47
83	8	17

47 In Fig. 20 the temperature increase has been plotted against the time interval between blowing. The B.t.u. loss per hour was computed from the mean temperature increase and is also shown in the same figure for ratings of 150 and 200 per cent.

TABLE 7 DATA AND RESULTS FOR OPERATING TESTS

Test No.	Duration, hr.	Per cent rating	Coal fired, lb. per hr.	Water fed to boiler, lb. per hr.	Actual evap. per lb. coal as fired	Superheated steam pressure, lb gage	Temperature, deg. fahr.				Factor of evaporation	Equiv. evap. per lb. coal as fired	B.t.u. per lb. coal as fired	Per cent boiler efficiency	Barometer in Hg. corr.	Boiler-room temp., deg. fahr.
							Steam	Superheat	Feedwater	Flue-gas potentiometer						
		Flue-	Blowing	Runs:												
80	40	154.8	12392	104106	8.40	225	607	210	172.5	543	1.213	10.20	12554	78.1	29.42	88
81	36	155.8	12000	104900	8.74	223	601	204	173.0	543	1.211	10.58	12663	81.1	29.36	90
82	24	156.4	12180	104435	8.58	225	622	224	171.0	558	1.222	10.48	12649	80.3	29.43	87
83	20	156.6	12679	103496	8.16	225	646	249	171.4	554	1.235	10.07	12340	79.2	29.50	86
		Totals	of Dead-	Bank Per	iods:											
	117[1]	55.2	5519	43813	7.95	220	616	221	172.0	—	1.218	9.68	12450	75.4	—	—
		Totals of	Floating-	Bank Per	iods:											
	94[2]	65.3	5450	43800	8.06	222	607	209	175.0	—	1.210	9.75	12480	75.8	—	—
		Variable-	Rating	Runs:												
101	48	132.7	10624	88893	8.37	223	611	214	168.5	533	1.218	10.19	12611	78.4	29.50	81
102	48	73.5	6194	49548	8.00	223	595	198	168.0	544	1.210	9.68	12274	76.5	29.47	88
103	25	129.7	10580	86481	8.18	225	626	228	173.0	522	1.223	10.01	12375	78.7	29.38	87

[1] Dead-bank total includes 4 bank periods = 69.5 hr.

[2] Floating-bank total includes 3 bank periods = 64 hr.

COAL FOR BANKING

48 The banking test included the period from August 13 to August 22. From August 13 to 18, the first period, dead banks were carried at night and over Sunday, and from the 18th to the 22d, the second period, floating banks were carried nights and Sunday. During the daytime the boiler was steamed at about 157 per cent rating. These two periods were divided up as shown in Table 11, which gives the data and results obtained. In computing the coal chargeable to banking it was assumed that all water was evaporated at the efficiency corresponding to the average rating during the steaming period. Although part of the water was evaporated at lower ratings during the building up and burning down of the fire, the resulting lower efficiencies during these periods are considered as chargeable to banking. These tests were conducted while the boiler was baffled according to arrangement "C," so the efficiencies during the steaming periods are taken from the efficiency curve in Fig. 11. The results of these tests have been plotted in Fig. 21 and straight lines have been drawn through the points representing the two methods of banking. It seems probable that a straight line does not represent the true relation between the two variables, but the lack of sufficient data makes the determination of the exact relationship impossible.

VARIABLE-RATING TESTS

49 These tests were conducted while the boiler was baffled according to arrangement "C." Table 7 gives the test data and results. During these tests the boiler was steaming during the day according to plant practice, taking its share of the load fluctuations. At night both steaming and banking were tried with the following results:

Test	Average rating, per cent	Per cent efficiency	Per cent efficiency, tests 70–76	Diff. in efficiency, per cent
101 — Low steaming at night	132.7	78.4	82.4	4.0
102 — Dead bank at night	73.5	76.5	81.6	5.1
103 — Steaming at night	129.7	78.7	82.5	3.8

During test No. 101 the boiler was steamed at night at a rating corresponding to nine boilers carrying the load, that is, at about

60 per cent rating. During test No. 102 it was banked from 5.00 p.m. to 6.00 a.m., and during test No. 103 it was steaming at night as one of the seven boilers carrying the plant load. That the average rating for test No. 101 is greater than for test No. 103 is due to the fact that the ratio of low steaming hours to total hours of the test was greater in test No. 103 than in test No. 101.

50 For comparison the efficiency at constant rating from tests Nos. 70–76 is also given. The two tests with low-rate steaming

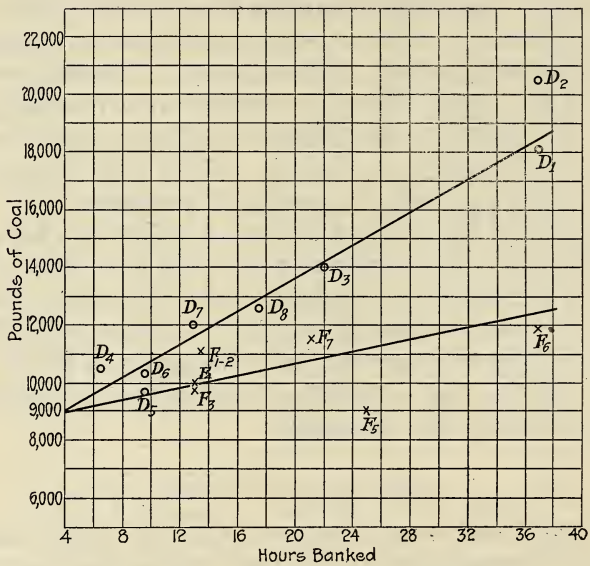

Fig. 21 Coal Required for Banking

show practically the same decrease below the efficiency at constant rating (4 per cent and 3.8 per cent), whereas the test during which a dead bank was carried at night shows a decrease of 5.1 per cent.

STARTING UP AND BURNING OUT FIRES

51 The weights of coal and water during starting-up and burning-out periods are given in Table 8, together with the resulting loss or gain. When burning out the fires the coal was run entirely out of the spreaders and stoker hoppers, so that this amount of coal had to be added to or subtracted from the weighed coal over the starting or burning-out period. The spreaders held 14,400 lb. of coal and the stoker hoppers 9,000 lb. After test No. 66 both hoppers

TABLE 8 LOSS AND GAIN DUE TO STARTING UP AND BURNING OUT FIRES

Starting Up Fires								Burning Out Fires							
Period	%Rating[1]	Water, lb.		Coal, lb.				Period	%Rating[1]	Water, lb.		Coal, lb.			
		Weighed	Evap.[2]	Equiv.[3]	Weighed	Burned [4]	Loss [5]			Weighed	Evap.[2]	Equiv.[3]	Weighed	Burned [4]	Gain [6]
S_3	133	89151	126151	15100	57008	48008	32908	B_{10}	268	255632	218632	27000	1056	10056	16944
S_{11}	79	369595	406595	47700	67177	58177	10477	B_{30}	151	268255	231255	27600	6256	15256	12344
S_{40}	154	1006080	1043080	127000	169145	160145	33145	B_{33}	217	191442	154442	19100	879	9879	9221
S_{60}	196	488837	525387	64600	103850	94850	30250	B_{54}	226	285872	248872	33700	4280	13280	20420
S_{70}	190	549497	586497	67000	138920	115520	48500	B_{66}	213	347286	310286	39800	0	23400	16400
S_{73}	105	514945	551945	62000	99080	90080	28080	....	...	.....	.....	.....	....	.....	.....

[1] Is rating for run preceding or following the period.
[2] Weighed water ± change of water in boiler.
[3] Pounds of coal equivalent to pounds of water evaporated.
[4] Weighed coal ± coal in spreader.
[5] Burned − equivalent coal.
[6] Equivalent − burned coal.

and spreaders were emptied and 23,400 lb. had to be added; in all other cases 9,000 lb. was used. The weighed water was corrected for the reduction in density of water in a cold boiler and a steaming boiler, and for the variation in water level in the outer top drums. Both these items have been estimated at 37,000 lb. This constant value has been used in computing the results shown in Table 8.

52 The loss and gain has been plotted against the per cent rating in Fig. 22. The ratings are those carried on the boiler at the

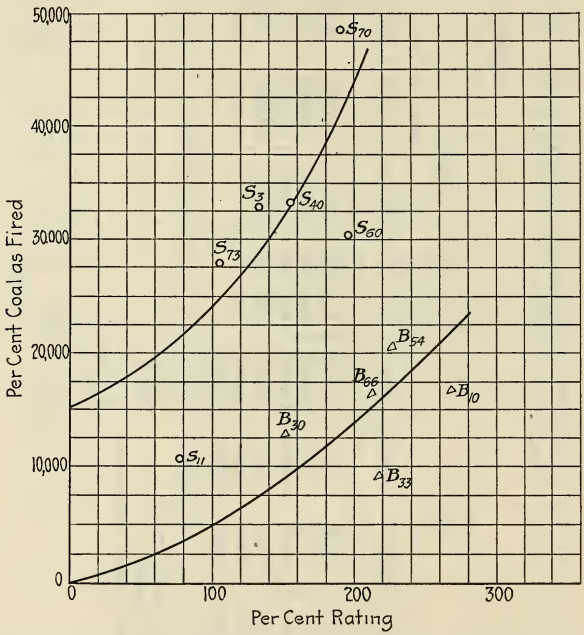

Fig. 22 Coal Lost or Gained in Starting Up and Burning Out Fires

end of the starting-up period and the beginning of the burning-out period.

CONCLUSIONS

53 *Baffling Arrangements.* The "A" baffling arrangement showed no improvement over the original arrangement except that the draft at the damper was reduced about 33 per cent.

54 With the "B" arrangement an average increase in superheat of over 20 deg. fahr. was obtained and the flue-gas temperature

was reduced about 70 deg. fahr. as compared with the original baffling. The draft was also about 38 per cent less than with the original baffling. The results obtained indicated that there was very little difference between the "B" and "C" arrangements and on account of the simplicity of the "B" arrangement and the fact that a slightly higher superheat was obtained, it was decided upon as the most practical arrangement to install in the present boilers.

55 *Extension Grates.* The changes made to the extension grates did not give the desired effect. After both the first and second changes the combustible present in the refuse was greater than with the old-type grates, and resulted from the trouble experienced with down blast which heated the grinders. It is believed that further study will develop a better arrangement of air openings, which will make possible a reduction in ashpit loss.

56 *Air Supplied for Combustion.* The method employed for measuring the air required for combustion was unsatisfactory, due to air leakage from the air chamber beneath the stoker, although the air passing through the meter was correctly indicated. The leakage from the air chamber amounted to as much as 25 per cent at high ratings.

57 *Comparison of Coals.* Shipper "B" coal gave only slightly better efficiency than Shipper "A" coal, while Shipper "C" coal was about 1 per cent and Shipper "D" coal 2.5 per cent lower than Shipper "A" coal. The results obtained, however, are not directly comparable due to the different arrangements of boiler baffling and conditions of the heating surface. A comparison on the basis of combustible lost in the refuse seems to be a better method. Expressed in percentage of dry coal the loss was:

Shipper "B"	1.6 per cent
Shipper "A"	2.25 per cent
Shipper "C"	3.68 per cent
Shipper "D"	4.25 per cent

Considerable difficulty was experienced with Shipper "D" coal due to the low fusion temperature of the ash, resulting in large clinkers, so that this coal was considered unsatisfactory. The other three burned satisfactorily, so that a consideration of cost f.o.b. plant, together with the relative value as fuel, determined the most economical coal. The results in this case were slightly in favor of Shipper "B."

58 *Energy for Auxiliaries.* The total energy consumed by the boiler auxiliaries averaged about 0.7 per cent between 170 per cent and 200 per cent rating, while at ratings near 50 per cent the energy consumed was 1.6 per cent. These figures are based upon the complete utilization of the exhaust steam in the feedwater heaters.

59 *Flue Blowing.* Taking into consideration the labor and maintenance costs directly influenced by the frequency of blowing flues together with the quantity of steam required per blow, an economic balance is obtained at 12.5 hr., assuming a constant rating of 200 per cent. Boilers banked at night should be blown only once a day. The foregoing is a literal interpretation of the test results which does not indicate exactly the proper procedure but serves more as a check on present practice.

60 *Banking.* Expressed in equation form, the coal required for the two methods of banking is:

$$\text{For floating bank, lb. coal} = 8600 + 105x$$
$$\text{For dead bank, lb. coal} = 8000 + 280x$$

where x = number of hours banked. These equations give a greater amount of coal for a floating bank than for a dead bank for periods of less than four hours; but under actual conditions there is practically no difference between the two kinds of banks for the first portion of a banking period, so the equations are meant to apply only to banking periods in excess of four hours.

61 *General.* At the present writing seventeen boilers have been rebaffled according to arrangement "B," modified slightly in the arrangement of the cross-baffle at the top of the front baffle. The vertical baffle has been shortened about six inches and the cross-baffles at the top inclined upward at about 35 deg. from the front baffle. The results being obtained indicate that the superheated-steam temperature has been increased fully 50 deg. fahr. over that obtained with the original baffle, and that there is a decided improvement in the overall efficiency.

62 A study of the heat absorbed in the different passes of the boiler indicated that a rearrangement of tubes would be beneficial. This possibility was studied by the engineers of the Babcock and Wilcox Company and The Detroit Edison Company, with the result that the new boilers recently installed at the Marysville power house have in cross-section but five tubes between the upper

center drum and each mud drum and ten tubes between each mud drum and the upper drum directly above. The boiler is baffled similarly to arrangement "B," except that there are two vertical baffles in each rear bank which direct the gases in three vertical passes after they leave the superheater pass. Economy results are not yet available, but it is believed that a decided decrease in the stack-gas temperatures will be realized.

DISCUSSION

L. P. Breckenridge. I am glad to express my appreciation of the fine work which this paper represents. It represents great care and accuracy. I cannot help but think back to the time when I began to make boiler tests. We used to make a lot of preparation and then make but one test. When I think of the development of the art of boiler testing as it has progressed during the last forty years it seems that this is a fitting climax, and is a representation of the type of work that we shall in the future expect to have reported to this Society.

Benjamin F. Grote. When one comes to a test which really goes into the merits of the performance of any kind of machine, it is something to think about. It is not a simple matter. It involves an immense amount of detail. The writer remembers one series of tests that he performed on a hydraulic turbine plant on which 40,000 instrumental readings were taken, and when we finished we found that we did not have enough to tell us what we wanted to know. Therefore, the writer believes that tests of this kind should be published in full.

D. S. Jacobus. It gives me great pleasure to discuss Mr. Thompson's paper, as it brings back the tests to which he refers that I made over ten years ago. As we look back the mellowing influence of time brings even more appreciation of kindness extended and there is no work that I remember with more pleasure than that at the Delray Plant where we made similar tests.

Few know of the magnitude of the undertaking when it comes to conducting tests of the sort. Perhaps a better comprehension would come to many if they knew the total cost of the tests. To conduct such a series of tests speaks well for the organization.

Shortsightedness often leads to not worrying about losses that cannot be seen and in unduly exaggerating expenses which can be seen. There are many plants that are losing money in a steady silent stream where it would pay to go to a lot of expense to find and apply a remedy. In the present case all the more credit is due, as the efficiency before the tests were projected stood out as a shining example of what could be accomplished, and to improve this efficiency so that the cost of the tests will be paid for many times over by the coal saving is surely an accomplishment.

My results are consistent with those obtained by Mr. Thompson. Mr. William Kent pointed out in his discussion of my paper [1] that my tests gave the highest efficiencies at high as well as low rates of driving ever obtained from coal having over 25 per cent of volatile matter, and it is indeed gratifying to note that the results are substantiated and that Mr. Thompson secured somewhat higher efficiencies than those found in my tests. The efficiencies secured by Mr. Thompson at the lower ratings with the baffles arranged in the same way as in my tests were the same as those which I obtained. At the higher ratings his results were about 2 per cent higher than mine, this difference, as shown by the heat balances, being due mainly to there being a lesser loss through the carbon in the ashes in his test than in mine. When corrected for the differences in the ashpit losses our results agree within one per cent, which is remarkably close when one considers the error that may be involved in estimating the condition of the fires at the beginning and end of a test with about 15 tons of coal on the grates.

The tests clearly indicate the advantage of changing the baffles. In addition to obtaining higher efficiencies the superheat was increased in the arrangement adopted for use and as a higher superheat could be made good use of this was also an advantage.

Tables 9 to 13 give the details which are necessary in making a study of the results. Among other details the draft losses are important in the study of a boiler of the sort where the stack effect of the furnace is a material factor.

I visited the plant when Mr. Thompson's tests were in progress and can certify to the fact that they were conducted in a most exact and scientific way. I have held up these tests on several occasions as an example of the sort of industrial research that

[1] Tests of Large Boilers at the Detroit Edison Company, Trans., vol. 33, p. 565.

should be encouraged by Engineering Foundation and the National Research Council, as they are a direct benefit in the problem of the conservation of fuel and to the engineer at large.

THE AUTHOR. The data and results of the banking tests mentioned in Par. 21 are shown in Table 13, which, with Tables 9 to 12, appear on the following pages.

Tables 4, 9 and 10 give the average data for all constant rating tests. The air quantities obtained by means of the air meter are shown in Table 9, and the air per pound of coal computed from the gas analysis in Table 10.

Table 12 gives the electrical energy required per hour for the blowers, stokers, grinders and the steam equivalent to the electrical energy, and the total steam for all auxiliary uses expressed as equivalent steam from and at 212 deg. fahr. The net efficiency is also given, deductions having been made for auxiliary steam used.

Under Table 2 are given the combined averages for all constant rating tests (3 to 76). During these tests a total of 20,570,093 lb. of coal was burned and the average efficiency obtained was 77.9 per cent. The average efficiencies obtained during these series of constant rating tests are indicated on Figs. 7, 8, 9, 10 and 11, respectively.

It should be appreciated that space prevents giving a complete report of the tests which form the subject of this paper. In order that complete data and results may be available for those who are particularly interested in this work, the author will supply a copy of the complete detailed report to the U. E. S. Library.

TABLE 9 TEST DATA FOR CONSTANT-RATING TESTS

Test No.	Date, Start & Finish		Duration, hr.	Per cent Rating	Coal		Water		Ash	Steam Press.		Temperatures								Drafts, in. water from oper. gages				Air & Vapor, lb. per hr. Thomas air meter	Vapor, Pounds per hr.	Barometer	Remarks
					Total	Hourly	Total	Hourly	lb.	Superh.	Satur.	Steam		Feedwater	Gas at Damper		Air at Duct	Outside	Room	Pressure Under Fire	Over Fire	At Damper					
												E.	W.		E.	W.						E.	W.				
2	4/8	4/10	32	166	297621	9301	2480667	77521	28300	226	230	566	600	149.3	524	569	70	44	69	0.56	0.010	0.22	0.25	113775	878	29.30	Fire burned out to fix
53			3		57008	19003	89151	29717	3177																		draft and Orsat lines.
3	4/11	4/12	39	133	415823	10684	3477587	89169	46400	224	230	586	602	159.8	577	597	77	50	73	0.79	0.012	0.33	0.34	134400	724	29.51	Burning-out period not
3–4			6		75838	12639	587361	97893	8298	225		586	590	159.0													recorded.
4	4/13	4/14	27	148.5	315500	11685	2695362	99828	34530	222	229	592	597	166.0	605	596	79	51	73	1.26	0.010	0.46	0.36	170150	1128	29.29	Run 4 — Opened East
4–5			7		93423	13346	767867	109695	15860	223	232	605	603	152.0	620	606											Damper one tooth on
5	4/14	4/15	32	163.7	412238	12892	3529355	110292	41140	222	231	599	597	168.6	633	613	75	54	75	1.39	0.014	0.57	0.38	189978	1542	29.19	quadrant.
6	4/15	4/17	33	182	491608	14897	4020635	121837	52840	223	233	617	610	169.5	655	647	69	37	74	1.86	0.026	0.78	0.55	225553	1223	29.27	
7	4/17	4/18	38	199	612771	16123	5015107	131976	60880	223	235	619	622	161.8	695	674	73	39	73	1.96	0.031	0.82	0.60	233300	1230	29.35	
8A	4/18	4/19	15	215	272990	18199	2154613	143700	26700	227	238	625	628	173.6	716	721	74	49	76	2.70	0.028	0.98	0.78	262169	1516	29.40	
8B	4/19	4/20	24	210	426780	17741	3350316	139600	41620	223	237	632	637	171.8	709	731	79	58	80	2.90	0.026	1.16	0.95	289833	1890	29.28	
9	4/20	4/21	19	241	382347	20124	3036284	159805	36971	224	239	633	636	173.1	722		81	63	80	3.44	0.037	1.17	1.00	298900	2459	29.18	
9–10			2		37319	18660	257597	128798	3467	224	232	627	633	163.0	702												Run 10 * Average for
10	4/21	4/21	12	268	270503	22549	2119837	176653	25133	225	242	637	643	167.0	782	782	77	65	81	4.50	0.071	1.32	1.19	340764*	3281	29.08	first 3 hr. only. □In-
B10			7		1056	151	255632 □	36519	2700 ○																		cludes7600#blowdown.
3–10			264	178	3865204	14648	31356704	118780	399716	223		615	615	167.0	663	673											○ Estimated 2700#.
																											Fire burned out to
S11			7		67177	9597	369595 □	5228	3700 ○	206	204	548	542	165.0	482	486											wash tubes.
11B	4/26	4/27	38	79	244724	6440	2081754	54783	24100	222	225	531	524	173.3	466	480	82	66	81	0.82	0.004	0.20	0.13	101400	1085	29.25	Run S11 ○ Estimated
11–12			3		12417	4139	98580	32860	1500	220	224	532	522	137.0													3700#. □ Includes
12	4/28	4/29	40	102.8	332377	8308	2856077	71402	40360	222	226	543	533	171.5	486	502	71	57	73	1.10	0.005	0.28	0.17	123700	981	29.36	10000# blow-down.
12–13			4		14259	3565	175402	43850	1760	222	225	524	522	177.0	470	466											P. 11–12 boiler banked
13	4/29	5/2	56	53.2	241396	4311	2071594	37000	31900	221	223	523	521	173.7	447	447	65	48	73	0.29	0.001	0.11	0.07	66248	360	29.39	to repair feedpump.

13–14			2		18965	9482	156726	78363	1614	224	230	540	530	173.0	510	516												
14	5/2	5/3	36	126.7	343900	9553	3133840	87051	29500	224	229	547	537	170.1	501	508	68	47	72	0.76	0.003	0.26	0.16	116300	830	29.36		
15	5/3	5/5	36	154.5	426269	11850	3818135	106000	44200	222	231	566	554	173.5	544	549	76	55	74	1.58	0.015	0.47	0.30	177114	1314	29.38	Run 15 — Ran out of	
24–16			2		28450	14225	231396	115698	2815	226	230	573	563	166.0	585	595											Shipper "A" coal, start-	
16	5/13	5/14	26	180	371060	14280	3210842	123480	36725	225	234	584	577	179.6	613	601	78	57	81	2.70	0.035	0.64	0.44	236500	1858	29.28	ed on shipper "B" (run	
16–17			6		30580	5097	294783	49296	3386	223	225	565	561	178.0	491	503											20).	
17	5/14	5/17	66	67.4	347966	5270	3068282	46480	36033	223	224	553	549	174.6	474	464		50	74	0.41	0.005	0.13	0.08			29.52	P. 24–16 blew front	
17–18			3		38110	12703	343002	114334	3614	225	237	575	571	176.0	551	561											banks with hand lance.	
18	5/17	5/18	24	216.7	403038	16793	3532668	147194	37000	228	239	591	594	176.8	650	641	73	60	78	2.80	0.046	0.83	0.64	255000	2133	29.47	Run 18 — Mixture of	
11–15			215	99.0	1634303	7600	14392008	66940	181634	222		544	536	171.5	495	505											Shipper "A" & "C"	
16–18			127	122.8	1219204	9600	10680975	84102	119133	226		576	572	176.5	580	570											Creek Coal during last	
																											8 hr. of run.	
15–20			5		44410	8882	395898	79180	4470	222	227	561	557	174.0	517	523												
20	5/5	5/7	36	108.4	298916	8300	2680922	74500	30170	223	227	564	553	176.0	509	511	73	57	76	1.08	0.003	0.28	0.18	127720	924	29.33		
20–21			3		32066	10689	302291	100764	3181	226	233	568	558	183.0	534	544												
21	5/7	5/8	36	153.3	411726	11440	3693847	102700	37900	224	231	577	568	173.7	578	567	73	58	76	1.68	0.010	0.49	0.33	188691	1181	29.34	Run 21 — Coal sample	
21–22			2		29190	14545	244060	122030	2545	225	235	588	582	174.0	598	606											from hopper lost.	
22	5/8	5/10	30	193.8	436161	14539	3955685	131836	37750	225	236	588	584	174.3	613	610	79	61	81	2.26	0.027	0.67	0.48	215300	1495	29.36	Scale sample used. Run 24 — Check on run	
22–23																											21.	
23	5/10	5/11	24	224.9	410675	17100	3645271	151800	28325	227	240	609	607	176.9	699	682	78	64	83	3.40	0.048	0.95	0.70	273640	2080	29.25	B30 boiler down to re-	
23–24			14		156491	11180	1279201	91372	20030	225	229	594	584	179.0	574	598											pair grinders.	
24	5/11	5/13	32	145.4	340545	10640	3176774	99200	30250	224	229	588	572	175.4	597	563	75	59	76	1.34	0.019	0.43	0.27	162900	1573	29.19	S31 ○ Total ash from	
24–16			14		144502	14225	1374912	115698	12100	224	228	566	554	167.0	571	547											end of run 30 to start	
20–24			196	154.5	2304682	11758	20748860	105862	206220	225		577	569	175.4	580	596											of run 31. Coal for start-	
																											ing fire not recorded.	
18–30			4		52356	13089	421940	105485	5330	227	233	593	585	174.0													Run 32 — Rating reduc-	
30	5/18	5/20	32	151.4	396056	12380	3312508	103516	58162	225	230	588	580	180.6	560	560	75	61	78	1.94	0.019	0.47	0.33	190000	1783	29.53	ed due to hot grinders.	
B30			6½		6256	962	268255	41270																			Run33 — ○ Ash includes	
S31			2		21347	10673	66740	33370	12000 ○																		burning-out period.	
31	5/21	5/22	33	115.0	310503	9409	2587189	78400	46280	224	228	575	572	173.5	535	524	82	79	89	0.86	0.007	0.28	0.18	125200	1400	29.44	B33 ○ See run 33.	
31–32			14		124141	8867	1090652	77189	19640	224	229	579	575	175.0	532	520											Boiler down for baffle	
32	5/23	5/24	16	184.1	250924	15683	1986298	124200	28350	226	234	606	601	170.6	626	612	83	71	90	2.66	0.065	0.68	0.49	221600	2650	29.33	change and repair.	
32–33			8		74297	9262	577075	72134	17203	223	228	596	592	176.0	551	541											P. 31–33 does not in-	
33–33	5/24	5/25	24	216.5	432239	18010	3503315	145971	61500 ○	225	237	616	616	174.7	658	652	79	70	84	3.65	0.101	1.01	0.78	280600	3638	29.27	clude starting-up and	
B33			4		879	220	191442	47860																			burning-out.	

TABLE 9 TEST DATA FOR CONSTANT-RATING TESTS (*Continued*)

Test No.	Date, Start & Finish		Duration, Hr.	Per cent Rating	Coal		Water		Ash	Steam Press.		Temperatures									Drafts, in. Water from Oper. Gages				Air & Vapor, lb. per hr. Thomas air meter	Vapor, Pounds per hr.	Barometer	Remarks
												Steam		Feedwater	Gas at Damper		Air to Duct	Outside	Room	Pressure Under Fire	Over Fire	At Damper						
					Total	Hourly	Total	Hourly	lb.	Superh.	Satur.	E.	W.		E.	W.						E.	W.				
31–33			95	149.8	1192104	12580	9744529	102500	172793	225		603	599	175.5	591	581											
S40			12		169145	14095	1006080	83840	12000 ○	224	227	589	569	179.0	572	560											S40 ○ ash for starting-up period estimated.
40	5/31	6/1	28	153.5	357633	12773	2932381	104728	40280	223	230	603	582	176.4	620	608	72	66	78	1.51	0.004	0.33	0.22	195000	1820	29.41	
40–41			4		58543	14633	473140	118285	6290	225	233	608	592	177.0	672	652											
41	6/1	6/3	31	199.0	505435	16300	4167003	134420	54070	226	233	615	603	178.0	700	681	79	68	83	2.88	0.053	0.52	0.46	253431	2728	29.39	P. 42–43 blew front banks.
41–42			4		51659	12915	446166	111516	10940	221	226	615	603		663	641											
42	6/3	6/4	17	225.8	323389	19023	2577721	151630	29240	226	239	622	606	180.7	738	713	76	69	83	3.23	0.078	0.71	0.59	280500	3192	29.39	Run 43 — Mixture of Shipper "A" & "D" coal during last 8 hr.
42–43			6		46540	7756	383710	63818	15530	224	229	571	589	179.0													
43	6/4	6/5	26	108.0	231993	8922	1917467	73749	23470	224	231	582	564	177.2	582	550	61	53	67	0.58	0.0022	0.22	0.13	142500	767	29.64	
40–43			116	164.3	1575192 △	13579	12897588	111186	179820 △	225		605	592	177.0	660	644											P.40–43 △ totals for all runs & periods between, not including starting-up period.
50	6/5	6/6	35	106.1	318712	9106	2539495	72557	41500	223	228	582	560	178.5	589	554	67	60	73	1.00	0.002	0.24	0.15	148650	1012	29.63	
50–51			4		27766	6941	228154	57038	3285																		
51	6/6	6/8	32	137	377374	11793	2992607	93519	45080	223	232	586	562	175.5	627	581	71	61	74	1.16	0.014	0.30	0.19	167395	1392	29.49	Run 54 — stopped on account of back fire in hopper & hot grinder.
51–52			2		21618	10809	199180	99590	2645	225	230	593	575	174.0	630	594											
52	6/8	6/9	30	163.6	418720	13957	3306106	110200	53370	226	232	601	588	171.6	675	646	79	69	81	2.03	0.033	0.40	0.31	227000	2248	29.44	
52–53			4		26799	6700	189970	47492	8460	222	227	587	579		658	628											B54 fires burned out for baffle change.
53	6/9	6/10	23	194.3	371476	16149	3026547	131589	37780	225	239	596	593	178.4	697	666	84	77	89	2.10	0.063	0.49	0.40	241603	3066	29.38	
53–54			1		19425	19425	136945	136945	2280	229	236	599	583	178.0	730	700											50–54 does not include burning-out.
54	6/10	6/11	17	226.4	351006	20647	2594073	152592	41130	225	243	622	595	175.4	751	751	82	73	89	3.40	0.090	0.77	0.53	299800	4466	29.23	
B54			6		4280	713	285872	47645	25520																		P.65–66 high rating lowered on account of hot grinders. Run was stopped to clean fires.
50–54			148	151.1	1932896	13100	15213077	103000	235530	225		598	580	175.7	674	642											
S60			6		103850	17290	488837	81473	8960	198	200	608	578	172.0	612												

60	6/29	6/30	32	196.4	519268	16227	4228013	132125	63380	227	237	628	598	173.4	612	590	80	77	90	2.89	0.104	0.43	0.32	228500	3329	29.21	B.66 Fires burned out
60–61			1		9339	9339	99854	99854	1166	227	233	591	581	176.0	647	635											for baffle change.
61	6/30	7/2	36	168.4	501427	13929	4080781	113355	62714	225	232	613	622	177.8	560	557	82	77	90	1.73	0.013	0.26	0.29	189426	2621	29.24	Hoppers & spreader
61–62			4		43108	10777	379264	94816	4450	224	227	605	615	174.0	530	528											emptied.
62	7/2	7/3	28	142.0	314665	11238	2666019	95215	32500	227	233	608	619	174.7	526	525	86	85	95	0.95	0.083	0.21	0.22	124000	1695	29.24	○ Estimated 4000#.
62–63			9		89126	9903	817236	90804	9255	225	231	614	620	178.0	537	527											S70 △ Includes 5000#
63	7/3	7/5	41	106.2	337008	8219	295951	72072	36335	226	232	600	595	176.9	471	461	86	83	93	0.37	0.018	0.11	0.11	85500	1312	29.36	blow-down.
63–64			2		7787	3893	105702	52851	1660	224	226	609	607	178.0													B72 Blew front bank
64	7/5	7/6	11	59.0	50832	4621	441152	40105	7504	227	227	578	587	172.4	407	407	86	81	95	0.07	0.007	0.05	0.05	57522	859	29.37	with hand lance.
64–65			3		21582	7194	168822	56274	2546	225	227	593	591	180.0													Burned out fire to
65	7/6	7/7	32	148.5	381193	11912	3175867	99246	42080	227	233	622	627	173.4	527	526	87	85	97	1.43	0.051	0.18	0.20	159600	2644	29.37	repair leaky baffle.
65–66			18		260528	14473	1940197	107788	36620	225	233	623	629	176.0	557	561											□ Aux. feed lines open
66	7/8	7/8	12	212.5	217340	18112	1690783	140899	18433	228	240	638	646	171.2	617	622	85	81	96	3.40	0.25	0.5	0.55	252675	3786	29.25	the last few hours
B66			7				347286	49612	4000 ○	226	228	603	611	169.0													while burning out,
60–66			242	144.8	2857053	11806	23584904	97460	331600	226		613	615	175.4	546	542											water fed through
																											them estimated at
S70			6		138920	25153	549497 △	549497	7480	222	226	555	559	161.0	515	521											50000#.
70	7/25	7/26	32	190.0	469998	14687	4106479	128327	43740	224	233	596	602	174.8	544	550		83	95	2.39	0.164	0.42	0.44			29.47	S73 ○ Total ash
70–71			6		46855	11714	367496	91874	11580	226	231	596	600	166.0													from end of run 72
71	7/26	7/28	36	162.0	459834	12773	3944753	109575	59760	227	230	594	596	175.0	517	527	105	83	93	1.59	0.150	0.34	0.36			29.31	to start of 73.
71–72			2		18768	9384	178030	89015	2180	223	228	580	582	177.0	494	500											Run 76 —Trouble with
72	7/28	7/30	38	134.0	405902	10682	3462027	91106	47280	223	227	592	591	175.1	497	501	106	77	88	1.03	0.098	0.24	0.24			29.18	hot grinders, load low-
B72			3				247000 □	82333																			ered for some time.
S73			9		99080	11009	514945	59216	17420 ○	224	227	548	552	156.0													B76 Estimated at
73	8/1	8/2	36	105.3	291517	8098	2590329	71953	33220	223	225	561	565	167.1	424	424	88	65	76	0.42	0.045	0.13	0.14			29.36	7000#.
73–74			5		53793	10759	472752	94550	5550	225	230	578	580	179.0													P.70–76 — △ Total for
74	8/2	8/4	35	149.0	414278	11836	3510012	100286	45950	225	231	616	617	171.0	507	506	96	67	79	1.65	0.098	0.30	0.30			29.42	all runs and periods in-
74–75			4		53077	13269	429341	107335	5560	226	232	619	623	163.0	524	514											cluding two starting-
75	8/4	8/5	27	183.2	395261	14639	3283315	121604	41920	232	237	629	635	166.5	567	549	92	74	87	2.10	0.180	0.46	0.47			29.43	up and burning-out
75–76			7		114055	16293	843586	120512	14560	232	238	610	614	172.0													periods.
76			18	208.5	310621	17257	2496054	138669	41800	225	235	625	638	172.0	575	556	90	74	87	3.22	0.377	0.74	0.78			29.16	73–76 Total for all runs
B76			4				314194	78548	7000 ○																		and periods including
73–76			145	150.6	1731682 △	11940	14454528 △	101070	213080 △	227		612	610	169.0	550	534											one starting-up and
70–76			266	153.8	3271959 △	12300	27309810 △	103350	385100 △	226		601	603	171.5	536	530											one burning-out period.

TABLE 10 COAL AND FLUE GAS ANALYSIS

Test No.	Per cent rating	Flue Gas Analyses								Lb. air per lb. coal at damper	Lb. air per lb. coal at superheater	Theoretical air per lb. of coal	Excess air coefficient at damper	Coal Analyses										Combustible in refuse
		At damper				At superheater								Proximate dry				Moisture %	Ultimate dry					
		CO_2	O_2	CO	N_2	CO_2	O_2	CO	N_2					F.C.	Vol.	Ash	B.t.u.		C	H_2	O_2	N_2	S_2	
2	116.0	13.0	5.8	0.11	81.1	13.6	4.8	0.15	81.4	12.89	12.33	9.46	1.363	56.60	32.10	11.30	13107	3.15	72.58	5.18	8.70	1.22	1.02	13.10
3	133.0	12.3	6.5	0.13	81.1	12.9	6.1	0.13	80.9	13.32	12.68	9.47	1.407	54.90	33.60	11.50	13233	4.37	72.16	5.49	8.35	1.22	1.28	13.20
4	148.5	12.6	6.2	0.15	81.1	14.2	4.6	0.16	81.0	13.15	11.67	9.26	1.420	55.30	33.50	11.20	13133	4.92	73.55	4.44	8.76	1.30	0.75	14.40
5	163.7	13.2	5.6	0.13	81.1	14.2	4.5	0.16	81.1	12.54	11.60	9.46	1.326	54.30	34.80	10.90	13374	4.70	72.82	5.34	8.46	1.38	1.10	13.10
6	182.0	13.2	5.8	0.10	80.9	14.1	4.8	0.16	80.9	12.49	11.63	9.43	1.325	54.40	34.10	11.50	13197	5.42	74.26	4.79	7.12	1.33	1.00	19.20
7	199.0	14.0	4.9	0.19	80.9	15.0	3.7	0.18	81.1	11.53	10.87	9.30	1.240	55.45	31.95	12.60	13054	5.95	73.07	4.95	6.59	1.50	1.29	13.80
8A	215.0	13.8	5.4	0.16	80.6	14.2	4.5	0.15	81.1	11.76	11.45	9.36	1.255	55.85	32.45	11.70	13254	6.20	74.12	4.84	6.90	1.35	1.09	18.70
8B	210.0	12.7	6.4	0.16	80.7	13.6	5.3	0.14	81.0	12.90	12.10	9.47	1.360	55.85	32.45	11.70	13254	4.97	74.12	4.84	6.90	1.35	1.09	18.70
9	241.0	14.5	4.7	0.14	80.7	15.2	2.9	0.18	81.7	11.40	11.02	9.35	1.220	55.15	34.55	10.30	13365	6.10	74.59	4.82	7.85	1.54	0.90	14.65
10	268.0	14.6	3.7	0.28	81.4					11.48		9.37	1.225	56.15	34.50	9.35	13257	5.40	74.19	4.97	9.26	1.38	0.85	10.25
11B	79.0	10.2	9.4	0.09	80.3	10.4	8.9	0.09	80.6	16.35	16.11	9.44	1.732	57.35	33.25	9.40	13471	5.06	74.49	4.97	8.90	1.46	0.78	14.75
12	102.8	11.1	8.3	0.09	80.5	11.4	7.6	0.07	80.9	14.55	14.35	9.42	1.548	56.40	31.90	11.70	13311	5.62	73.98	5.08	7.04	1.20	1.00	19.40
13	53.2	8.7	11.2	0.07	80.0	9.0	10.6	0.06	80.3	18.92	18.20	9.49	1.992	56.40	32.47	11.12	13378	4.10	72.87	5.25	8.59	1.21	0.95	16.15
14	126.2	14.1	5.0	0.09	80.8	14.8	4.0	0.14	81.1	11.94	11.40	9.42	1.265	54.90	34.85	10.25	13352	5.79	75.12	4.72	7.45	1.32	1.14	14.15
15	154.5	13.3	5.8	0.10	80.8	14.3	4.6	0.12	81.0	12.44	11.38	9.55	1.305	54.60	31.45	13.95	13238	4.04	73.24	4.91	5.79	1.22	0.89	19.65
16	180.0	13.5	5.6	0.06	80.8	14.6	4.3	0.11	81.0	12.20	11.25	9.18	1.330	55.75	31.85	12.40	12868	3.46	72.15	4.45	8.13	1.57	1.30	15.45
17	67.4	10.2	9.6	0.09	80.1					16.09		9.32	1.727	55.88	32.27	11.85	13218	3.68	72.57	4.68	8.37	1.45	1.08	13.80
18	216.7	14.0	4.9	0.09	81.0	15.3	3.7	0.11	80.9	11.88	10.82	9.33	1.273	56.75	31.50	11.75	13110	3.31	73.07	4.49	8.59	1.20	0.90	18.00
20	108.4	11.5	7.8	0.07	80.6	12.1	7.2	0.12	80.6	15.00	14.19	9.72	1.542	58.50	31.95	9.55	13710	4.40	76.21	4.94	6.94	1.17	1.19	16.20
21	153.3	12.5	6.7	0.10	80.7	13.2	6.0	0.12	80.7	14.05	13.20	9.62	1.462	58.55	31.85	9.60	13832	4.00	76.21	4.36	7.42	1.21	1.20	12.30
22	193.8	14.6	4.3	0.14	81.0	15.1	3.7	0.22	81.0	11.51	11.04	9.60	1.199	56.85	33.00	10.15	13786	4.10	73.29	5.53	8.90	1.13	1.00	13.70
23	224.9	15.1	3.6	0.16	81.1	15.3	3.2	0.27	81.2	11.52	11.27	9.79	1.177	57.35	32.50	10.15	13741	2.90	74.95	4.99	7.73	1.12	1.06	12.50
24	145.4	13.5	5.6	0.10	80.8	14.3	4.8	0.13	80.8	13.00	12.26	9.84	1.321	58.75	33.50	7.75	13927	2.79	75.83	5.08	9.06	1.25	1.03	16.65
30	151.4	13.1	6.1	0.11	80.7	13.7	5.4	0.09	80.8	12.20	11.68	9.17	1.330	57.50	30.10	12.40	12853	4.25	71.74	4.52	9.29	1.18	0.87	20.05
31	115.0	13.2	5.9	0.05	80.9	14.1	4.8	0.09	81.0	12.21	11.37	9.20	1.327	55.95	30.90	13.15	12694	3.24	71.41	4.76	8.95	1.22	0.51	20.65
32	184.1	14.3	4.7	0.07	80.9	14.7	4.2	0.06	81.0	11.40	11.52	9.08	1.255	57.10	30.30	12.60	12760	3.59	71.57	4.45	9.45	1.21	0.72	15.05
33	216.5	14.0	5.0	0.07	80.9	14.6	4.2	0.10	81.1	11.60	11.15	9.15	1.260	56.70	30.00	13.30	12763	2.98	72.25	4.10	7.94	1.38	1.03	21.40

40	153.5	12.5	6.7	0.07	80.7	13.9	5.1	0.08	80.9	12.90	11.68	9.26	1.394	55.80	31.15	13.05	12847	4.70	72.08	4.73	7.03	1.15	1.96	18.00
41	199.0	13.0	6.1	0.06	80.8	14.9	4.1	0.09	80.9	12.55	10.95	9.42	1.332	56.90	31.55	11.55	13136	4.30	72.89	5.02	8.10	1.42	1.02	19.25
42	225.8	13.7	5.5	0.10	80.7	15.6	3.1	0.11	81.2	12.50	11.03	9.70	1.290	55.85	33.35	10.80	13351	4.03	72.76	4.32	5.98	1.22	0.92	23.20
43	108.0	10.2	9.1	0.07	80.4	11.8	7.4	0.10	80.7	15.45	13.35	9.12	1.745	55.45	31.40	13.22	12935	4.47	71.60	4.72	7.42	1.30	1.23	21.60
50	106.1	9.6	10.0	0.06	80.3					15.20		8.70	1.747	52.05	32.75	15.20	12481	4.55	67.92	4.52	8.59	1.40	2.37	21.35
51	137.0	11.8	7.6	0.04	80.6					13.02		9.21	1.460	52.20	33.40	14.40	12603	4.21	69.03	4.60	7.97	1.44	2.56	17.55
52	163.6	12.3	6.9	0.05	80.7					13.04		9.26	1.408	54.00	33.00	13.00	12864	3.63	72.74	4.00	6.12	1.29	2.85	23.80
53	194.3	13.3	5.6	0.09	81.0					11.36		9.07	1.251	53.02	34.60	12.38	12881	3.24	70.14	4.65	8.70	1.35	2.78	23.80
54	226.4	13.6	5.0	0.14	81.2					10.90		9.00	1.210	54.65	31.95	13.40	12745	3.79	70.18	4.35	8.32	1.26	2.49	32.00
60	196.4	14.8	4.0	0.09	81.1	15.1	3.6	0.11	81.2	10.70	10.51	9.13	1.172	55.05	32.07	12.88	12785	4.50	71.85	4.63	8.39	1.34	0.91	22.75
61	168.4	14.7	4.2	0.15	81.0	15.1	3.6	0.10	81.2	10.99	10.75	9.12	1.203	55.40	32.45	12.15	12983	4.07	71.74	4.74	9.64	1.24	0.49	18.25
62	142.0	14.8	4.1	0.11	81.0	15.4	3.5	0.08	81.0	10.85	10.45	9.20	1.180	56.35	31.80	11.85	12877	4.95	71.55	5.17	9.13	1.35	0.95	15.30
63	106.2	14.3	4.6	0.10	81.0	15.0	4.2	0.09	80.7	11.32	10.75	9.09	1.247	57.95	29.35	12.70	12890	4.88	72.23	4.60	8.31	1.17	0.99	16.00
64	59.0	11.1	8.4	0.06	80.4	12.6	8.3	0.02	79.1	14.69	12.72	9.35	1.570	57.15	31.55	11.30	12988	3.38	71.98	5.09	9.29	1.33	1.01	18.00
65	148.5	15.1	3.6	0.12	81.2	15.4	3.3	0.16	81.1	10.42	10.14	9.08	1.148	55.85	31.12	13.03	12651	3.58	70.07	5.12	9.86	1.28	0.94	20.30
66	212.5	15.1	3.6	0.12	81.2					10.68		9.25	1.154	56.00	32.10	11.90	12840	3.96	73.68	4.23	8.06	1.36	0.77	29.30
70	190.0	14.1	4.6	0.11	81.2	15.2	3.6	0.12	81.1	11.85	10.97	9.52	1.245	54.62	33.22	12.15	13104	2.83	74.47	4.33	7.57	1.26	0.71	24.80
71	162.0	13.7	5.1	0.09	81.1	14.6	4.2	0.09	81.1	11.93	11.22	9.18	1.283	53.20	33.20	13.62	12810	3.65	72.40	4.18	7.70	1.28	1.12	17.50
72	134.0	13.7	5.1	0.08	81.1	14.7	4.3	0.10	80.9	11.72	10.88	8.98	1.275	54.10	32.60	13.30	12743	4.66	71.93	4.48	7.95	1.34	1.00	17.80
73	105.3	13.8	5.1	0.08	81.0	14.5	4.4	0.10	81.0	12.25	11.62	9.39	1.305	57.70	30.70	11.60	13018	3.60	74.26	4.30	7.62	1.36	0.86	16.70
74	149.0	13.7	5.4	0.07	80.8	14.4	4.6	0.07	80.9	12.05	11.51	9.18	1.313	55.60	31.90	12.50	12855	5.73	73.69	4.38	7.05	1.34	1.04	12.80
75	183.2	14.1	4.7	0.08	81.1	15.2	3.6	0.10	81.1	11.56	10.87	9.12	1.267	55.90	33.30	10.80	13149	5.47	73.68	4.36	8.61	1.52	1.03	18.30
76	208.5	14.3	4.7	0.10	80.9	15.4	3.4	0.11	81.1	11.37	10.57	9.10	1.250	55.60	32.80	11.60	13298	4.93	73.09	4.23	8.93	1.26	0.89	20.10
80														54.13	33.45	12.42	13115	4.28	73.12	4.48	7.70	1.37	0.91	
81														54.52	32.68	11.70	13188	3.98	74.70	4.83	6.49	1.34	0.94	
82														54.65	33.55	11.80	13222	4.33	73.98	5.06	6.96	1.46	0.94	
83														53.90	32.50	13.60	12829	3.76	71.74	4.29	7.96	1.40	1.01	
101														55.35	33.92	10.73	13214	4.56	74.11	4.72	8.34	1.26	0.84	
102														54.07	33.40	12.53	12880	4.66	72.37	4.54	8.32	1.38	0.86	
103														54.20	33.80	12.00	13016	4.85	74.15	4.63	7.13	1.27	0.82	

TABLE 11 REFUSE DATA

Test No.	Per cent rating	Pounds coal fired per hour	Lb. dry refuse per hr.—weighed	Weighed refuse in % coal as fired	Per cent ash in coal as fired	Per cent combustible in dry refuse	Per cent refuse computed	Combustible lost in refuse % coal as fired	Refuse lost to stack, % coal as fired	Refuse lost to stack, % of total computed	Fusing point of ash deg. fahr.
2	116.0	9301	822	8.94	10.92	13.10	12.58	1.66	3.64	29.00	2483+
3	133.0	10684	1190	11.15	11.00	13.20	12.67	1.67	1.52	12.00	2483+
4	148.5	11685	1278	10.95	10.20	14.40	11.92	1.72	0.97	8.13	2483+
5	163.7	12892	1285	9.98	10.38	13.10	11.95	1.57	1.97	16.48	2483+
6	182.0	14897	1600	10.75	10.90	19.20	13.49	2.59	2.74	20.30	2483+
7	199.0	16123	1602	9.94	11.85	13.80	13.75	1.90	3.81	27.70	2483+
8A	215.0	18199	1780	9.79	10.97	18.70	13.52	2.55	3.73	27.60	2483+
8B	210.0	17741	1735	9.77	11.30	18.70	13.89	2.59	4.12	29.60	2483+
9	241.0	20124	1945	9.73	9.70	14.65	11.35	1.65	1.62	14.25	2483+
10	268.0	22549	2090	9.27	8.85	10.25	9.86	1.01	0.59	6.00	2483+
3–10	178.0	14640	1517	10.35	10.79	15.20	12.73	1.94	2.38	18.70	
11B	79.0	6440	810	12.60	8.91	14.75	10.45	1.54			2477+
12	102.8	8309	1009	12.15	11.05	19.40	13.71	2.66	1.56	11.38	2477+
13	53.2	4311	570	13.22	10.50	16.15	12.53	2.03			2477+
14	126.2	9553	820	8.58	9.65	14.15	11.24	1.59	2.66	23.60	2477+
15	154.5	11850	1229	10.38	13.35	19.65	16.62	3.27	6.24	37.50	2477+
16	180.0	14280	1412	9.90	11.97	15.45	14.76	2.19	4.26	30.10	2477+
17	67.4	5270	546	10.36	11.41	13.80	13.24	1.83	2.88	21.75	2477+
18	216.7	16793	1545	9.19	11.35	18.00	13.84	2.49	4.65	33.60	2477+
11–15	99.0	7600	844	11.10	10.96	17.10	13.20	2.24	2.10	15.90	
16–18	122.8	9600	938	9.77	11.55	15.85	13.70	2.15	3.93	28.70	
20	108.4	8300	837	10.09	9.12	16.20	10.88	1.76	0.79	7.26	2489+
21	153.3	11440	1052	9.18	9.20	12.30	10.48	1.31	1.33	12.70	2489+
22	193.8	14539	1260	8.66	9.75	13.70	11.30	1.55	2.64	23.35	2489+
23	224.9	17100	1178	6.88	9.86	12.50	11.27	1.41	4.39	39.50	2489+
24	145.4	10640	943	8.85	7.53	16.65	9.03	1.50	0.18	0.02	2489+
20–24	154.5	11760	1052	8.96	9.23	14.10	10.75	1.52	1.79	16.75	
30	151.4	12380	1815	14.65	11.87	20.05	14.85	2.98	0.20	0.01	2483+
31	115.0	9409	1405	14.95	12.70	20.65	16.02	3.32	1.07	6.68	2483+
32	184.1	15683	1773	11.30	12.15	15.05	14.30	2.15	3.00	21.00	2483+
33	216.5	18010	2560	14.23	12.90	21.40	16.41	3.51	2.18	13.30	2483+
31–33	149.8	12550	1820	14.50	12.65	19.60	15.75	3.10	1.25	7.94	
40	153.5	12773	1440	11.28	12.44	18.00	15.17	2.73	3.89	25.65	2311
41	199.0	16300	1743	10.70	11.17	19.25	13.81	2.64	3.11	22.55	2483+
42	225.8	19023	1723	9.06	10.38	23.20	13.49	3.11	4.43	32.80	2483+
43	108.0	8922	903	10.12	12.65	21.60	16.14	3.49	5.12	31.65	2489+
40–43	164.3	13580	1550	11.43	11.57	20.23	14.50	2.93	3.07	21.20	
50	106.1	9106	1185	13.02	14.50	21.35	18.44	3.94	5.42	29.40	2214
51	137.0	11793	1410	11.97	13.80	17.55	16.74	2.94	4.77	28.50	2163
52	163.6	13957	1777	12.73	12.52	23.80	16.43	3.91	3.70	22.50	2136
53	194.3	16149	1640	10.15	13.55	23.80	17.78	4.23	7.63	42.40	2136
54	226.4	20647	2420	11.70	12.90	32.80	18.97	6.07	7.27	38.35	2214
50–54	151.1	13060	1590	12.20	13.40	23.60	17.55	4.16	5.35	30.50	
60	196.4	16227	1980	12.20	12.27	22.75	15.89	3.62	3.69	23.20	2483+
61	168.4	13929	1740	12.50	11.66	18.25	14.25	2.59	1.75	12.30	2483+
62	142.0	11238	1160	10.32	11.24	15.30	13.20	2.06	2.98	22.60	2483+
63	106.2	8219	885	10.76	12.10	16.00	14.40	2.30	3.64	25.25	2483+
64	59.0	4621	684	14.80	10.90	18.00	13.30	2.40			2483+
65	148.5	11912	1315	11.04	12.25	20.30	15.37	3.12	4.33	28.20	2483+
66	212.5	18112	1535	8.47	11.42	29.30	16.14	4.72	7.67	47.50	2483+
60–66	144.8	11806	1370	11.60	11.85	19.90	14.80	2.95	3.20	21.60	
70	190.0	14687	1368	9.32	11.81	24.80	15.69	3.88	6.37	40.60	2483+
71	162.0	12773	1660	13.00	13.10	17.50	15.88	2.78	2.88	18.15	2483+
72	134.0	10682	1243	11.65	12.68	17.80	15.42	2.74	3.77	24.45	2483+
73	105.3	8098	923	11.40	11.18	16.70	13.42	2.24	2.02	15.05	2483+
74	149.0	11836	1220	11.30	11.78	12.80	13.50	1.72	2.20	16.30	2483+
75	183.2	14639	1552	10.60	10.20	18.30	12.49	2.29	1.89	15.14	2483+
76	208.5	17257	2325	13.50	11.01	20.10	13.78	2.77	0.28	0.02	2483+
70–76	153.8	12300	1445	11.75	11.78	18.37	14.45	2.67	2.70	18.70	

TABLE 12 COMPUTED RESULTS

Test No.	Per cent rating	Equivalent evaporation from & at 212° fahr., lb. per hr.	Equi. evap. per lb. of coal as fired	Equi. evap. per sq. ft. sat. heating surface per hour	Factor of evap.	Degrees superheat	Lb. coal per hr. per sq. ft. projected grate area	Kw-hr. per hour		Steam from & at 212° F. equi. to Kw-hr., lb. per hr.	Total aux. steam from & at 212° F. lb. per hour	Combined efficiency, per cent	Net eff., per cent
								Blower	Stoker & Grinder				
2	116.0	94850	10.19	3.65	1.224	185	32.8	14.9	3.6	86.7	778	77.9	77.3
3	133.0	108600	10.20	4.17	1.219	197	37.6	21.6	3.1	115.9	827	78.3	77.7
4	148.5	121300	10.38	4.64	1.214	199	39.4	25.6	3.2	135.1	865	81.2	80.6
5	163.7	133600	10.37	5.12	1.212	202	45.4	29.0	3.4	152.0	900	78.9	78.4
6	182.0	148700	10.00	5.64	1.220	217	52.4	42.2	3.6	214.6	983	77.2	76.7
7	199.0	162300	10.10	6.16	1.231	225	56.8	44.3	4.1	227.0	1013	79.9	79.4
8A	215.0	175800	9.66	6.64	1.223	229	64.1	59.0	4.0	295.5	1103	75.4	74.9
8B	210.0	171500	9.65	6.52	1.228	239	62.4	83.8	3.8	411.0	1210	74.4	73.9
9	241.0	196100	9.75	7.39	1.227	238	70.8	102.0	5.0	502.0	1337	75.4	74.9
10	268.0	218900	9.72	8.21	1.239	243	79.3		7.2			75.1	
3–10	178.0		9.93		1.224							76.8	
11B	79.0	64200	9.99	2.53	1.172	131	22.7	14.0	2.6	77.8	729	75.8	74.9
12	102.8	84100	10.12	3.30	1.178	142	29.5	16.9	2.9	92.4	772	78.1	77.4
13	53.2	43200	10.07	1.71	1.168	126	15.2	11.6	1.9	58.6	658	76.0	74.8
14	126.2	103000	10.77	4.03	1.183	145	33.6	14.5	3.5	84.5	792	83.1	82.5
15	154.5	126000	10.62	4.89	1.188	164	41.7	29.0	4.2	155.7	897	80.3	79.7
16	180.0	147000	10.31	5.68	1.191	183	50.3	59.0	5.1	300.3	1071	80.5	79.9
17	67.4	54600	10.43	2.14	1.182	154	18.6	12.4	2.2	68.5	705	79.5	78.5
18	216.7	176700	10.50	6.77	1.201	194	59.2	65.2	5.6	332.0	1145	80.5	80.0
11–15	99.0		10.39		1.179							79.5	
16–18	122.8		10.43		1.191							79.7	
20	108.4	88500	10.65	3.42	1.187	162	29.2	16.9	2.8	92.4	778	79.0	78.3
21	153.3	122900	10.71	4.75	1.194	175	40.3	30.8	3.4	160.5	896	78.3	77.7
22	193.8	158200	10.88	6.08	1.200	189	51.2	45.4	4.8	235.5	1021	79.8	79.3
23	224.9	183500	10.73	7.00	1.209	210	60.2	87.1	6.3	438.0	1259	78.1	77.6
24	145.4	118500	11.15	4.56	1.196	183	37.5	23.3	4.5	130.5	859	80.0	79.4
20–24	154.5		10.73		1.199							78.8	
30	151.4	123500	9.98	4.76	1.193	187	43.5	34.9	4.8	184.8	921	78.7	78.1
31	115.0	93700	9.96	3.63	1.196	177	33.4	16.1	4.6	97.1	790	78.6	77.9
32	184.1	150300	9.59	5.75	1.210	206	55.2	55.6	6.0	288.8	1061	75.6	75.1
33	216.5	177500	9.85	6.73	1.215	219	63.4	100.0	7.2	503.5	1314	77.1	76.5
31–33	149.8		9.86		1.207							77.4	
40	153.5	125500	9.83	4.83	1.199	195	45.0	28.3	4.7	154.7	893	77 9	77.3
41	199.0	162700	9.94	6.17	1.209	211	57.4	67.3	5.6	342.0	1133	76.7	76.2
42	225.8	183000	9.63	6.95	1.208	216	67.0	88.4	5.7	441.5	1262	72.9	72.4
43	108.0	87800	9.85	3.39	1.190	173	31.4	17.2	3.5	97.1	781	77.3	76.6
40–43	164.3		9.85		1.205							76.3	
50	106.1	86750	9.51	3.34	1.194	174	32.1	15.3	3.5	88.2	770	77.4	76.7
51	137.0	111300	9.45	4.32	1.193	177	41.4	22.6	4.5	127.1	846	76.0	75.4
52	163.6	133500	9.57	5.10	1.212	196	49.1	46.0	5.3	240.5	989	75.0	74.5
53	194.3	158500	9.81	6.08	1.205	197	56.9	51.3	6.8	262.5	1049	76.0	75.5
54	226.4	185300	8.96	7.08	1.215	201	72.6	98.5	9.2	505.5	1328	70.9	70.4
50–54	151.1		9.45		1.200							75.0	
60	196.4	160700	9.88	6.09	1.214	215	57.2	62.2	5.1	315.5	1102	78.5	78.0
61	168.4	137500	9.88	5.22	1.212	220	49.0	32.3	4.4	172.0	926	77.0	76.5
62	142.0	115500	10.31	4.39	1.214	216	39.6	16.3	4.2	96.2	818	81.7	81.1
63	106.2	86750	10.55	3.32	1.203	200	29.0	12.0	3.4	72.3	754	83.5	82.8
64	59.0	48100	10.46	1.85	1.201	185	16.2	9.8	2.3	52.1	677	80.9	79.8
65	148.5	120000	10.18	4.57	1.222	227	42.0	23.1	4.4	129.0	858	80.8	80.2
66	212.5	173500	9.60	6.53	1.233	244	63.8	79.0	6.0	398.5	1200	75.8	75.3
60–66	144.8		10.04		1.215							79.1	
70	190.0	155000	10.55	5.93	1.208	202	51.8	41.8	5.8	223.0	1003	80.7	80.2
71	162.0	132000	10.34	5.05	1.205	197	45.0	25.8	5.0	144.5	892	81.3	80.8
72	134.0	109300	10.27	4.19	1.201	195	37.6	16.5	3.8	95.2	810	82.0	81.4
73	105.3	86000	10.62	3.35	1.195	166	28.4	12.2	4.0	76.0	757	82.2	81.5
74	149.0	122000	10.28	4.65	1.215	220	41.7	22.0	3.8	121.0	856	82.4	81.8
75	183.2	149500	10.21	5.59	1.230	232	51.5	39.1	4.9	206.4	974	79.8	79.3
76	208.5	170000	9.85	6.42	1.226	231	60.8	65.1	6.5	335.5	1134	75.6	75.1
70–76	153.8		10.13		1.213							79.3	

TABLE 13 DATA FOR BANKING TESTS

Test	Start	Finish	No. bank periods	Hours banked	Total hours	Average hours per bank period	Total coal in pounds	Total water in pounds	Heating value of coal, B.t.u.	Total coal reduced to standard	Total coal computed from water evap.	Cost of banking in pounds of coal	Cost per bank period	Boiler rating, steaming periods	Boiler rating, Entire run
							Floating Bank								
F_1	10 A.M.8/18	9 A.M.8/20	2	27	47	13.5	314027	2550707	12640	318000	295800	22200	11100	156.0	81.5
F_2	10 A.M.8/18	4 P.M.8/20	2	27	54	13.5	399957	3289831	12560	402200	380000	22200	11100	157.4	91.5
F_3	10 A.M.8/19	9 A.M.8/20	1	13	23	13	152020	1245163	12620	154000	144200	9800	9800	157.4	81.0
F_4	10 A.M.8/19	4 P.M.8/20	1	13	30	13	237950	1985237	12600	240000	230000	10000	10000	158.0	99.2
F_5	4 P.M.8/19	8 A.M.8/22	2	50	64	25	274050	2188195	12390	272100	254000	18100	9050	159.0	50.6
F_6	9 A.M.8/20	8 A.M.8/22	1	37	47	37	196682	1566090	12270	193400	181500	11900	11900	158.6	48.5
F_7	10 A.M.8/18	8 A.M.8/22	3	64	94	21.3	510709	4116797	12480	510709	476000	34709	11570	157.4	65.3
							Dead Bank								
D_1	1 P.M.8/13	9 A.M.8/15	1	37	44	37	98745	693000	12410	98450	80400	18050	18050	144	52.8
D_2	1 P.M.8/13	5 P.M.8/15	1	37	52	37	197470	1527798	12460	197500	177000	20500	20500	154.5	43.9
D_3	1 P.M.8/13	8 A.M.8/16	2	44	67	22	307080	2416920	12500	308100	280000	28100	14050	155.0	53.8
D_4	9 A.M.8/15	8 A.M.8/16	1	6.5	23	6.5	208335	1723490	12500	209100	198500	10600	10600	157.0	112.0
D_5	9 A.M.8/15	8 A.M.8/17	2	19.5	47	9.7	367065	3007184	12460	367500	348000	19500	9750	157.0	95.0
D_6	5 P.M.8/15	2 P.M.8/17	2	19.5	45	9.7	345997	2796306	12400	344100	323500	20600	10300	157.0	93.2
D_7	10 A.M.8/16	10 A.M.8/18	2	26	48	13	313329	2494574	12400	312600	288500	24100	12050	156.0	77.5
D_8	1 P.M.8/13	10 A.M.8/18	4	69.5	117	17.4	645694	5130000	12450	645694	595000	50694	12683	156.5	55.2

	Floating Bank	Dead Bank
Ave. boiler rating, steaming periods	156.8	156.5
Efficiency, steaming periods	81.9	81.9
Ave. boiler rating, entire tests	65.3	55.2
Efficiency, entire tests	75.75	75.4
Heating value of standard coal, B.t.u.	12480	12450

No. 1873

FEED HEATING FOR HIGH THERMAL EFFICIENCY

ECONOMIES OF 25,000-KW. POWER STATIONS USING SINGLE- AND MULTIPLE-STAGE CONDENSER HEATERS, WITH AND WITHOUT ECONOMIZERS, DETERMINED TO DEMONSTRATE THE EFFECT OF VARYING THE FEEDWATER TEMPERATURE

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Junior Member of the Society

The author investigates the problem of determining the correct feedwater temperatures for conditions of high thermal efficiency. He applies his methods to a 25,000-kw. power station, investigating the conditions when using single- and multiple-stage condenser heaters, both with and without economizers. In appendices are given all the necessary data for the solution of the problems and the formulas by which the results are obtained.

The paper is illustrated with curves obtained from the solution of the problem, various conditions being considered and the results compared. The paper considers the problem from the viewpoint of thermal efficiency only and does not include other factors such as investment, operating cost, etc.

FOR power plants using single- or multiple-stage feedwater heaters of the condenser type, the temperature of the boiler feedwater as it leaves the heaters should not be less than 150 deg. fahr. when using economizers and, with the main unit operating on 330-lb. per sq. in. steam pressure, 200 deg. fahr. superheat, and 29 in. vacuum, probably not more than 260 deg. fahr. for two-stage heating when not using economizers. Three- and four-stage heating, however, permit improving thermal efficiencies up to temperatures of 300 to 340 deg. fahr., corresponding to pressures of 72 to 120 lb. per sq. in. absolute in the heater. The maximum of this range was established by consideration of fuel charges only. The lower limit of 150 deg. fahr. was chosen to avoid mechanical difficulties met when sending colder water to economizers, although this temperature, as will be seen, is less than the lowest temperature justified by purely thermal analyses of the 25,000-kw. plant used as a basis for the present studies.

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2 That there exists for any fixed set of conditions a definite feedwater temperature at which the efficiency of power generation is a maximum is most readily demonstrated by consideration of a theoretically perfect generating plant using perfect condenser heaters. Figs. 3 and 5 show that in this case, assuming that the quantity of condensate heated remains fixed, for both single- and double-stage heating, the weight of the exhaust steam required to heat the condensate is practically directly proportional to the rise in the temperature of the feedwater. If an open type of feedwater heater were used, maintaining a constant back pressure on the units supplying the exhaust steam, the load carried by these units also would

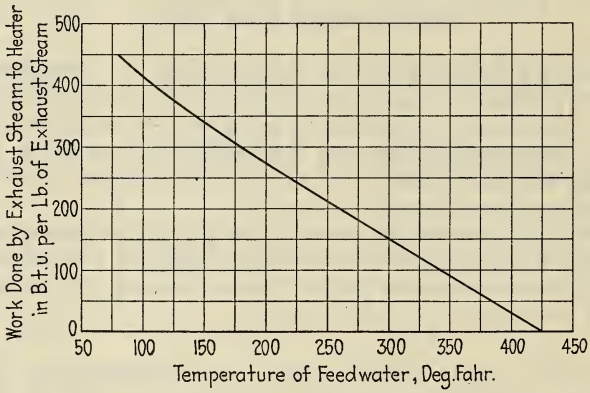

Fig. 1 Theoretical Work Done by Each Pound of Steam Used for Heating Water

(The expansion is adiabatic from the boiler pressure to that corresponding to the vapor tension of the feedwater. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.)

be approximately directly proportional to the rise in the temperature of the feedwater. However, since the feedwater heaters are considered as being perfect condensers, the pressure within them, and for the purposes of the theoretical analyses, the back pressure on the auxiliary turbine or at the extraction point on the main unit supplying the heating steam will be that corresponding to the vapor tension of the feedwater, and so will increase as the temperature of the feedwater increases. This increase in pressure in turn will increase the water rates of the steam units exhausting to the heaters, or, from another viewpoint, will decrease the work capable of being done by a pound of steam used for heating the feedwater. This is shown by the curve of Fig. 1.

3 Consequent upon increasing feedwater temperatures, therefore, we have a reduction in the amount of work obtainable from

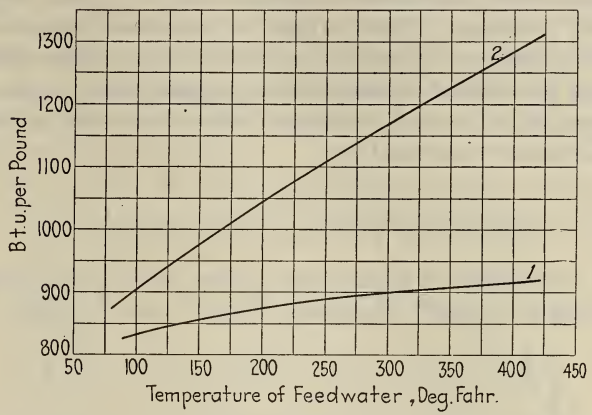

FIG. 2 THEORETICAL HEAT AVAILABLE FOR RAISING THE TEMPERATURE OF THE BOILER FEEDWATER

(Curve 1 is determined by subtracting from the heat content of the exhaust steam the heat content of the feedwater. The heat content of the exhaust steam, curve 2, in turn, is determined by subtracting from the initial heat content of the steam the heat equivalent of the work done by it expanding adiabatically to the pressure in the feedwater heater. Initial steam pressure, 325 lb. superheat, 200 deg. fahr.)

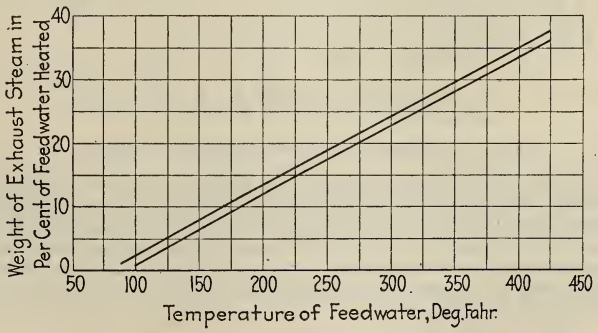

FIG. 3 WEIGHT OF EXHAUST STEAM REQUIRED

(The upper curve shows for single-stage heating the theoretical weight of exhaust steam required to heat the boiler feed from 79 deg. fahr., corresponding to a vacuum of 29 in. in the condenser of the main unit. The lower curve shows that required when the condensate temperature is 92 deg., corresponding to a vacuum of 28.5 in. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.)

each pound of steam used for heating the feedwater and an increase in the total amount of steam required for heating purposes. The relation between these offsetting effects is such that the work done

by the steam used for heating the feedwater increases as the temperature is increased to a certain point, after which it decreases. This is also shown by curves of Figs. 4 and 5 for both single- and double-stage heating. Any increase in feedwater temperature beyond that for which maximum work obtains continues to augment the demand for exhaust steam, but the capacity of a given quantity of the steam to do work is so reduced that the result is diminution in the total power generated by it.

EFFECT OF MULTIPLE-STAGE HEATING

4 A conception of what happens may be had by reference to the temperature-entropy diagrams, Figs. 6 and 7, given for both

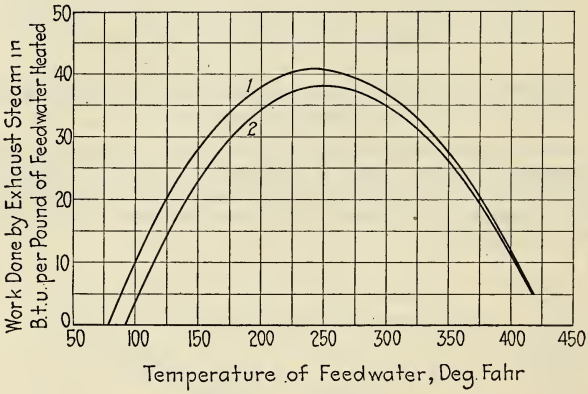

FIG. 4 WORK DONE BY EXHAUST STEAM

(Curve 1 shows for single-stage heating that the theoretical maximum work obtained from steam subsequently used for heating boiler feedwater is obtained with a final feedwater temperature of approximately 240 deg. fahr. when the initial temperature of the feedwater is 79 deg. fahr. Curve 2 is based on a condensate temperature of 92 deg. and shows that in this case the maximum work is obtained with the final feedwater temperature at 260 deg. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr. The peaks of the curves establish the feedwater temperatures for maximum theoretical thermal efficiency.)

single- and double-stage heating, assuming for simplicity that saturated instead of superheated steam is used. For convenience in illustrating the effect of varying the feedwater temperature, the areas representing the work done by the house turbine were placed above the water line of the temperature-entropy diagrams of the main unit. They should be viewed as distinct from these diagrams, however, though the scales used in either case are the same. It is evident that as the temperature of the feedwater T_f is increased

from T_c along AB the area *abcd*, which represents the work done by the house turbine for single-stage heating, at some point becomes a maximum. This is also true for the sum of the corresponding areas for double-stage heating. The temperatures at which these maxima occur are those for the best theoretical efficiency. Lowering either the initial or back pressure lowers the feedwater temperature for maximum efficiency, while increasing either of these pressures raises this temperature.

5 As the number of stages of feedwater heating increases, the work derived from the steam used for heating the feedwater increases

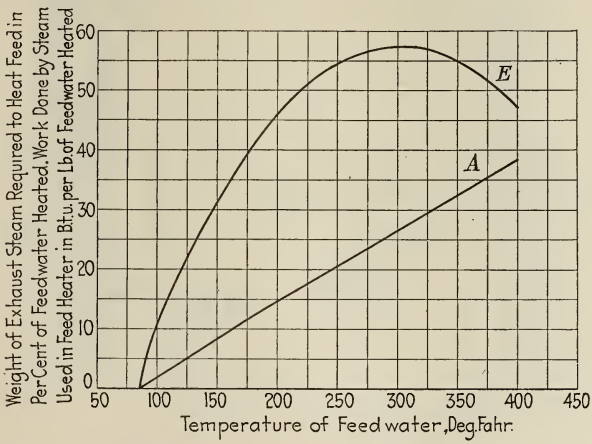

FIG. 5 THEORETICAL WEIGHT OF EXHAUST STEAM TO HEAT FEEDWATER AND TOTAL WORK DONE BY THIS STEAM

(With two-stage heating the theoretical weight of exhaust steam required to heat the feed water follows the curve marked A, while the total work done by this steam is given by curve marked E. Maximum work is derived from the exhaust steam at a feedwater temperature of approximately 300 deg., which theoretically, therefore, is the temperature for best thermal efficiency. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.; vacuum, 29 in.)

and the temperature of the feedwater, as established for maximum theoretical efficiency, approaches that of the initial steam. Using an infinite number of stages, the temperature of the feedwater for best efficiency is equal to that of the initial steam and the efficiency of the theoretical power-generating cycle is that of Carnot's cycle. This is shown on the temperature-entropy diagram for infinite-stage heating.

DESCRIPTION OF ASSUMED POWER STATION

6 Illustrative of what is involved in the practical problem of determining the most efficient feedwater temperatures for power

stations, the effect of varying this temperature was determined for several assumed stations of 25,000 kw. capacity, using various methods of heating the feedwater. To illustrate the influence of factors such as the water rate of the main unit and the slope of the Willans line of this unit, the internal Rankine-cycle efficiency of the bled steam and the Rankine-cycle efficiencies of the house turbine, two cases — designated Case 1 and Case 2, respectively —

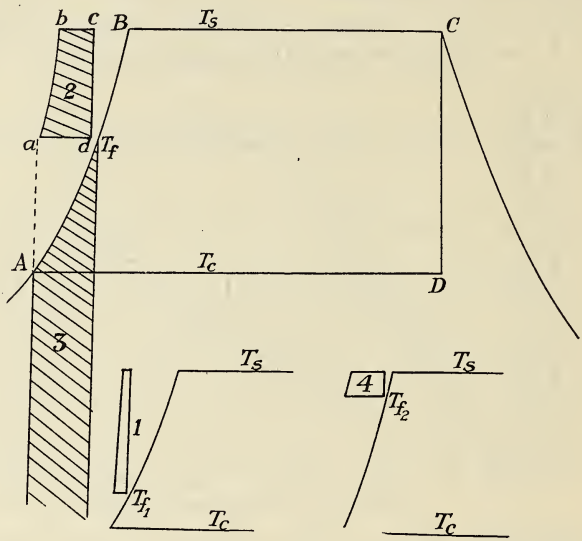

FIG. 6 TEMPERATURE-ENTROPY DIAGRAM FOR SINGLE-STAGE HEATING USING DRY SATURATED STEAM

(Large area *ABCD* represents the work done by the steam passing through the main unit. The shaded area 2, above the water line, represents the work done, before entering the heater, by the steam used for heating the feedwater. The shaded area 3, below the water line, represents the heat added to the feedwater and is equivalent to the area under line *ad*. As indicated by areas 1 and 4 of the diagram at the bottom of the illustration, the area representing the work done by the steam used for heating the feedwater becomes rather small when the feedwater temperature deviates largely from that for best efficiency. The letters T_f, T_{f1} and T_{f2} indicate the feedwater temperature for different positions of the area showing the work done by the steam used for heating the feedwater. T_S is the temperature of the initial steam and T_C is the temperature of the condensate.)

were worked out for each arrangement of feedwater heating. The Rankine-cycle efficiency used in Case 1 for the bled steam based on the steam pressure on the turbine side of the throttle was approximately 67 per cent, the slope of the Willans line was 12 lb. per kw-hr., and the water rate of the main unit when carrying the total gross station load was 10.6 lb. per kw-hr. For Case 2 the

corresponding values were 80 per cent for the Rankine-cycle efficiency of the bled steam, 9 lb. per kw-hr. for the slope of the Willans line, and 10.26 lb. per kw-hr. for the water rate of the main unit.

7 The Rankine-cycle efficiencies of the house turbines are given in Figs. 11 and 14. The efficiencies for the 650-kw. house

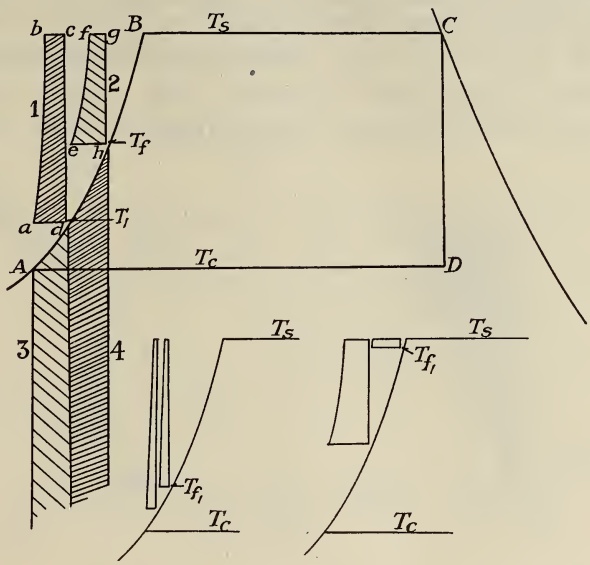

FIG. 7 TEMPERATURE-ENTROPY DIAGRAM FOR DOUBLE-STAGE HEATING USING DRY SATURATED STEAM

(Large area *ABCD* represents the work done by the steam passing through the main unit Instead of a single area above the water line, as for single-stage heating, two areas, *abcd* and *efgh* represent the work done by the steam passing respectively to the first and second stages. For equivalent heating effects in each heater, areas 3 and 4, below the water line, are equal. The figures at the bottom of diagram show that when the feedwater temperature is rather close to either the condensate temperature or the temperature of the boiler steam, the sum of the areas representing the work done by the steam used for heating the feedwater is small. When the temperature of the feedwater equals that of the boiler steam the effect is simply that of a single. stage heater. Evidently at some temperature between that of the condensate and that of the boiler steam the sum of the areas representing the work done by the steam used for heating the feedwater is a maximum. The temperatures indicated as T_f, T_{f1} and T_{f2} are those of the feedwater.)

turbines used in Case 1 were taken from curves similar in shape to those shown in Fig. 11, but were otherwise practically the same as those used for Case 2. At low back pressure the Rankine-cycle-efficiency curve of commercial house turbines is likely to deviate from the curves shown, improving rather rapidly in the vicinity of 26 in. vacuum and falling off beyond 28 in. The lines for Case 2, however,

represent sufficiently accurately for the present purpose the efficiencies that may be expected of the better designs of house turbines, while the shape of the curve for Case 1 probably approximates what may be expected of house turbines designed primarily for high back pressure. An average value of 80 per cent was used for the Rankine-cycle efficiency of the bled steam on the main unit, though this will vary 2 or 3 per cent either way, depending on the conditions of bleeding and the design of the turbine.

8 All auxiliaries in these stations were considered as being motor-driven during normal operation, dual exciters and spare steam-driven boiler-feed pumps being provided to assure continuity

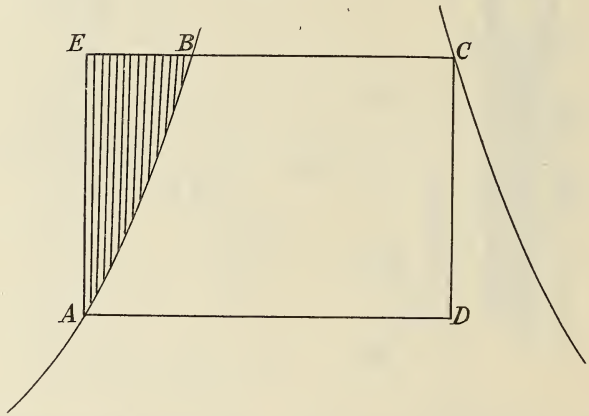

Fig. 8 Temperature-Entropy Diagram Using an Infinite Number of Stages

(The shaded area is the total work done by the steam used for heating the feedwater. The total work cycle, *AECD*, is seen to be a rectangle. The efficiency of this cycle is equivalent to Carnot's cycle.)

of operation in case of disruption of the electric service. As these steam units were for use only in emergencies they did not influence the various heat balances worked out. Surface condensers were used on the main units and the condensate before going to the feedwater heaters was passed as cooling water through the evaporator system supplying boiler-feed make-up water so long as this operated on exhaust steam. When the temperature of this exhaust steam was too low to evaporate the water efficiently and the use of live steam became necessary, the condensate was returned directly to the feedwater heater and the water used for the condenser of the evaporator was taken from the boiler feedwater previously heated

in the economizers or in the feedwater heaters. With this arrangement the use of live steam on the evaporators did not materially affect the heat balance, and the pressure of the exhaust steam used for heating the feedwater was determined by the pressure within the feedwater heater rather than the requirements of the evaporator system. A schematic arrangement of these plants is shown in Fig. 9. Economizers are shown, but heat balances as well were worked out for plants not using economizers, in which case the boiler feedwater was delivered directly to the boiler.

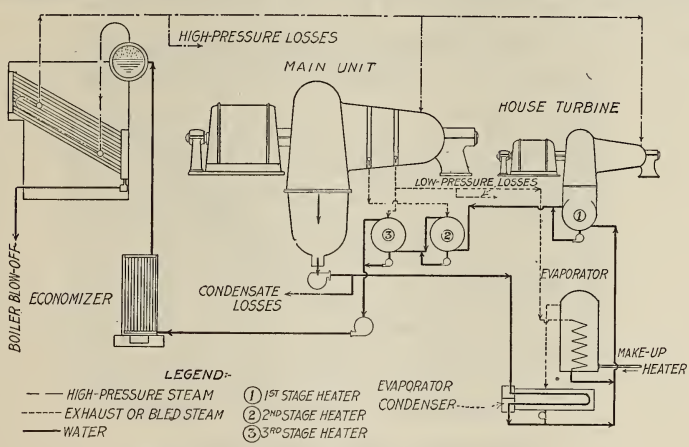

Fig. 9 Diagram of Station Layout

(Schematic diagram of station layout for three-stage feedwater heating, showing distribution of steam and the various sources of water and steam losses. The condensate from the main unit is circulated through the condenser of the make-up water evaporator system before going to the feedwater heater.)

9 Radiation losses from the exhaust-steam piping and losses from low-pressure traps were considered as being independent of the feedwater temperature, although additions to these losses and to the necessary boiler-feed make-up water were made to compensate for increasing leakages from the low-pressure system with increasing pressure in that system. The extent of these additions is illustrated by the dotted heat-consumption curves of case 2, and by the curve of make-up water. The quantities of steam lost from the low-pressure systems in any case can only be roughly estimated, although they should increase with the pressure in the heaters and therefore tend to lower the temperature for best econ-

omy. In this study the losses were considered as occurring by intermittent leakages, so that under usual operation the full amount of steam bled or supplied by the house turbines would be available for heating the feedwater. Radiation losses from the low-pressure system were taken as equivalent to 2 per cent of the heat in the low-pressure steam. The steam pressure at the boiler was taken at 350 lb. per sq. in. and that at the throttle of the turbine as 330 lb. per sq. in., the superheat in either case being taken as 200 deg. fahr. The terminal difference, that is, the difference between the temperature of saturated vapor at the pressure in the heater and

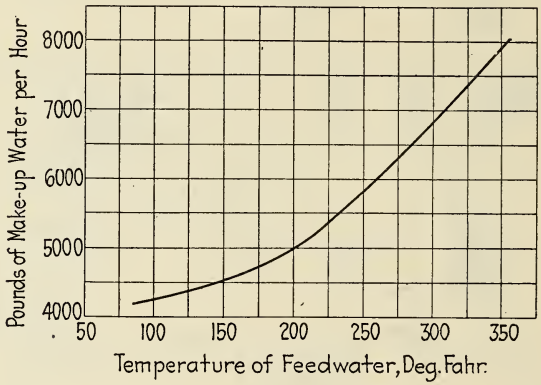

FIG. 10 BOILER-FEED MAKE-UP WATER

(The curve has an upward trend due to increases in the low-pressure steam losses at the higher pressures in the auxiliary steam piping.)

the temperature of the leaving boiler feedwater, was taken as 5 deg. fahr. The auxiliary power requirements were taken as 1300 kw., making the total load on the main unit when carrying the entire station 26,300 kw., at which load its steam consumption was 279,000 lb. per hr. for Case 1 and 270,000 lb. per hr. for Case 2. For every 100 kw. decrease in load on the main unit the steam consumed by it was reduced 1200 lb. per hr. for Case 1 and 900 lb. per hr. for Case 2. High-pressure drips were 1000 lb. per hr. and equivalent in heat content to 390,000 B.t.u. Condensate losses were 1000 lb. per hr. The heat content of the exhaust or bled steam was determined by the following formula:

$$\text{B.t.u. per lb. of exhaust or bled steam} = \frac{HW - 3415(L + K)}{W}$$

where —

H = heat content of initial steam, B.t.u.

W = weight of steam passing through house turbine or bled from the main unit, lb.

L = load on house turbine or carried by bled steam, kw.

K = factor for radiation, friction and generator losses expressed in kilowatts. For the house turbine, K was approximately 10 per cent of the full-load capacity of the house turbine used. For bled steam, the no-load losses were considered as being carried by the steam passing on to the condenser and K was equal only to the radiation losses and did not exceed 1 per cent of the power generated by the bled steam.

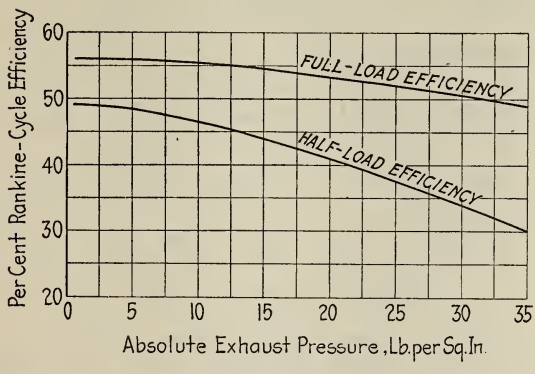

FIG. 11 RANKINE-CYCLE EFFICIENCIES FOR CASE 1

(Estimated Rankine-cycle efficiencies of 1500–1700-kw. house turbine generators as used for determining the heat balance for Case 1. The efficiencies used for the 650-kw. house turbine for Case 1 were approximately the same as those of Case 2, though the shape of its curve is similar to the ones given here. Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.)

When bleeding the main unit the drop in pressure in the bleeder piping varied with the amount of steam bled, and this was taken into consideration. The boilers were operated between 175 and 200 per cent of rating when economizers were not used, and under this condition had an efficiency of 78 per cent. When economizers were used the operating capacity was increased to 225 per cent of rating and the efficiency reduced to 75 per cent, not including the economizers. The power taken by the induced-draft fans was not included in the auxiliary load, and to obtain the true heat-consumption rate for the stations using economizers the equivalent heat consumption of these fans will have to be added to the rates that are given.

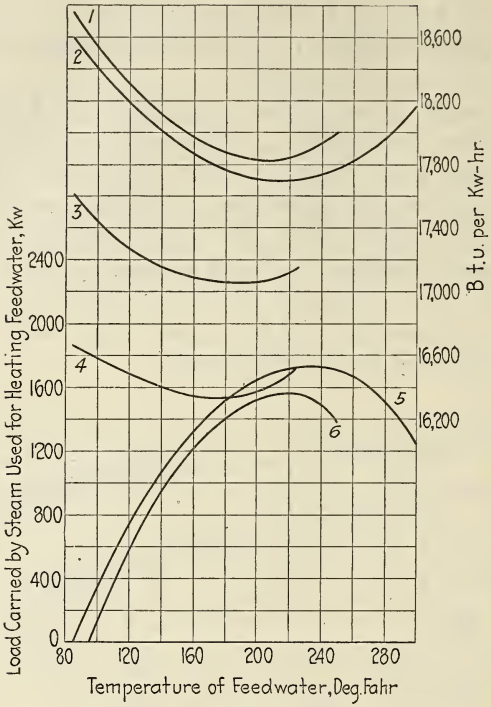

FIG. 12 SINGLE-STAGE HEATING — CASE 1

(Steam pressure, 330 lb. gage.; superheat, 200 deg. fahr.; vacuum on the main unit. 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1600-kw. house turbine and not bleeding the main unit Economizers were not used. The feedwater temperature for best efficiency is approximately 210 deg. fahr.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit. Economizers were not used. The feedwater temperature for best economy is slightly higher than for Curve 1.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area and bleeding the main unit. The economizer surface is approximately equivalent to 50 per cent of the boiler area. The feedwater temperature for best economy is approximately 190 deg. fahr. The heat consumption of the induced draft fans, equivalent to the power consumed by them, is not included.

CURVE 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface and bleeding the main unit. Though the economizer area is double that for Curve 3, the feedwater temperature for best economy is reduced only 10 to 15 deg. The power taken by the induced-draft fans was not included with the auxiliary power when determining the heat-consumption curves.

CURVE 5. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit. If low-pressure leakage losses are ignored the temperature for best economy when not using economizers would be determined by the peak of this curve, which occurs at 240 deg. fahr.

CURVE 6. Load developed by the 1600-kw. house turbine supplying exhaust steam to the feedwater heater. A comparison of the temperature at which the peak of Curve 6 occurs with that of Curve 5 indicates the influence that decreasing Rankine-cycle efficiencies with increasing

back pressures on the house turbine have on the feedwater temperature for best economy. If low-pressure steam leakages are ignored the feedwater temperature for best economy for Curve 6 would be 220 deg., or 20 deg. less than for Curve 5. By reference to Curves 1 and 2, showing the B.t.u. rates per kw-hr., it is seen that the real influence of the decreasing Rankine-cycle efficiencies with increasing back pressures is small.

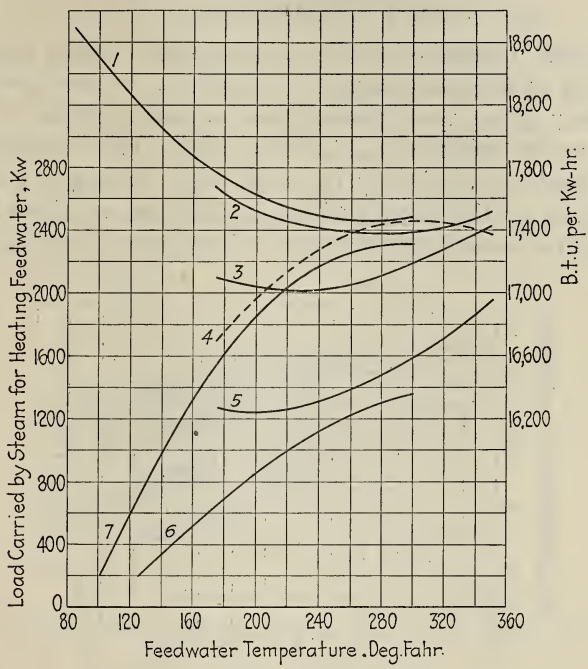

FIG. 13 DOUBLE-STAGE HEATING — CASE 1

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1400-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately 280 deg. fahr.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. The feedwater temperature for best economy is slightly higher than that shown by Curve 1, though, due to the flatness of the curves, it may be considered the same. It is noticeable that as the number of stages used for heating the feedwater increases, the curve of heat consumption per kw-hr. flattens out.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. The economizer surface is approximately equivalent to 45 per cent of the boiler area. The feedwater temperature for best efficiency is between 210 deg. and 250 deg. fahr. The power taken by the induced-draft fans was not included in the auxiliary load.

CURVE 4 (dotted). Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points. If low-pressure losses are ignored the temperature for best economy when not using economizers would be at the peak of this curve, or 300 deg. fahr.

CURVE 5. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit at two points. The feedwater temperature for best economy is between 180 and 220 deg. fahr., or 35 deg. less than shown by Curve 3.

Curve 7. Load developed by the steam used for heating the feedwater when using a 1400-kw. house turbine and bleeding the main unit only for the second-stage heating. Curve 6 shows the load carried by the house turbine for this arrangement of heating the feedwater.

HEAT BALANCE FOR SINGLE-STAGE HEATING

10 Heat balances for two arrangements using single-stage heating and no economizers were worked out. In one arrangement the auxiliary power was obtained from a house turbine of 650 kw. capacity, or one-half the total auxiliary load. The remainder of the auxiliary load was carried by the main unit. Steam, in addition to that supplied by the house turbine, in this case was bled from the main unit for heating the feedwater. The other arrangement used

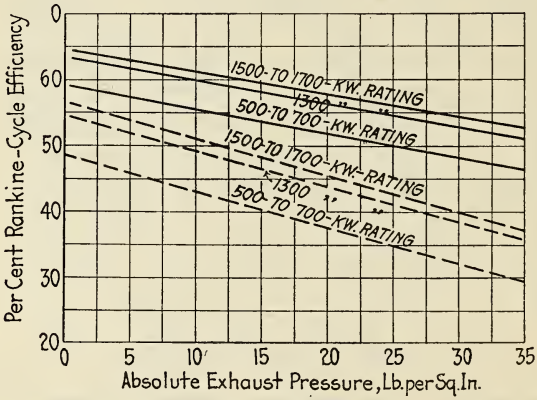

Fig. 14 Case 2. Estimated Rankine-Cycle Efficiencies of Turbine Generators Designed for Any of the Various Back Pressures and Operating against the Back Pressure for Which They Are Designed

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr. The efficiencies would deviate somewhat from those shown with back pressures below 2 lb. absolute.)

a house turbine with its point of best economy at 1600 kw. for Case 1 and 1700 kw. for Case 2, and was considered as being so designed that the house turbine could deliver power to the main bus. With this arrangement no means for bleeding the main unit were provided, all steam for heating feedwater being obtained from the house turbine. As indicated by the curves 1 and 2 of Figs. 12 and 15, the first arrangement is thermally the more economical and also requires for best efficiency a slightly higher feedwater temperature than the second arrangement. It is interesting to note that whereas the theoretical feed temperature for best efficiency is in

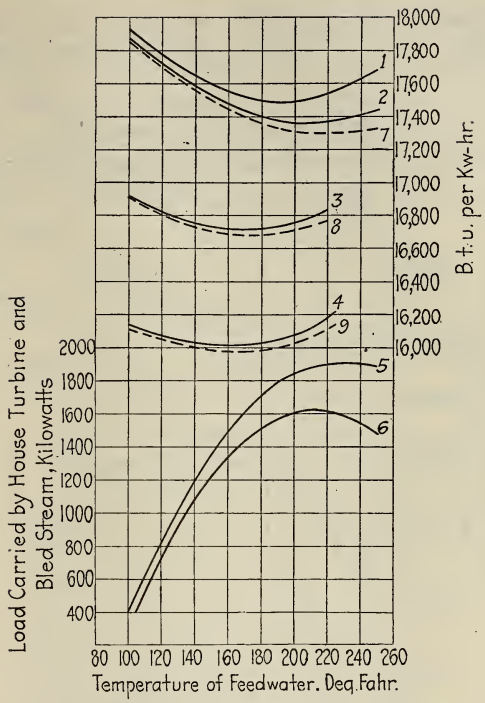

FIG. 15 SINGLE-STAGE HEATING — CASE 2

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1700-kw. house turbine and not bleeding the main unit. Economizers were not used. The feedwater temperature for best economy is approximately 190 deg. fahr. This is 20 deg. lower than the best feedwater temperature for the corresponding conditions of Case 1, though, due to the flatness of the curves over this range, the proper temperature for either case may be considered as approximately the same. The difference in temperature indicated is due to the difference in the slopes of the Rankine-cycle efficiency curves for the house turbine.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit, not using economizers. The feedwater temperature is slightly higher than for Curve 1.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area, and bleeding the main unit. The feedwater temperature for best economy is between 160 and 180 deg. fahr. This is lower than the corresponding temperature for Case 1, due largely to the difference between the slopes of the Willans lines of the main units. The power consumed by the induced-draft fans was not included in the auxiliary power.

CURVE 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface, and bleeding the main unit. The power consumed by the induced-draft fan was not included in the auxiliary power.

CURVE 5. Load developed by the steam used for heating the feedwater using a 650-kw. house turbine and bleeding the main unit.

CURVE 6. Load developed by the 1700-kw. house turbine supplying exhaust steam to the feedwater heater.

CURVES 7, 8 and 9 (dotted) are the same as 2, 3 and 4 respectively except that the low-pressure steam losses are not included.

the neighborhood of 250 deg. fahr., the actual temperature is closer to 200 deg. fahr. This is due to various factors, among them be-

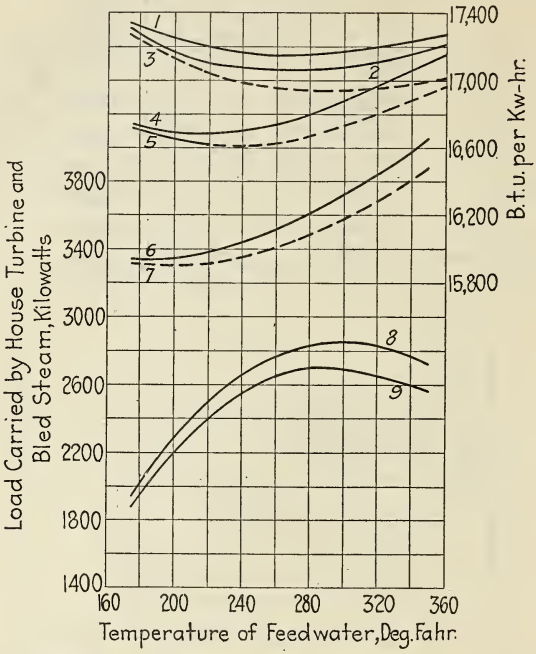

FIG. 16 DOUBLE-STAGE HEATING — CASE 2

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1500-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately the same as for the same conditions of Case 1.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. The feed temperature for best economy tends to be slightly higher than for Curve 1, though the difference is not appreciable. Curve 3 is the same except that losses due to leakage from the low-pressure steam piping are not included.

CURVE 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. The feedwater temperature for best economy is between 200 and 240 deg. fahr., or slightly lower than for Case 1. The power taken by the induced-draft fans was not included in the auxiliary power load. Curve 5 is the same but does not include the low-pressure steam losses.

CURVE 6. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit. The feedwater temperature for best economy is between 160 and 200 deg. fahr. This is lower than for Case 1, due to difference in the slope of the Willans lines of the main units. Curve 7 is the same but does not include the low-pressure steam losses.

CURVE 8. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

CURVE 9. Load developed by the steam used for heating the feedwater when using a 1500-kw. house turbine and bleeding the main unit. Fig. 18 shows the load carried by the house turbine.

ing the decreasing Rankine-cycle efficiency of house turbines when operating at successively higher back pressures.

HEAT BALANCE FOR DOUBLE-STAGE HEATING

11 Similarly heat balances for two arrangements using double-stage heating and no economizers were worked out on the basis that the heating effect was divided equally between the two stages. In one of these arrangements the first-stage heaters derived the steam for heating the feedwater from a house turbine with its point

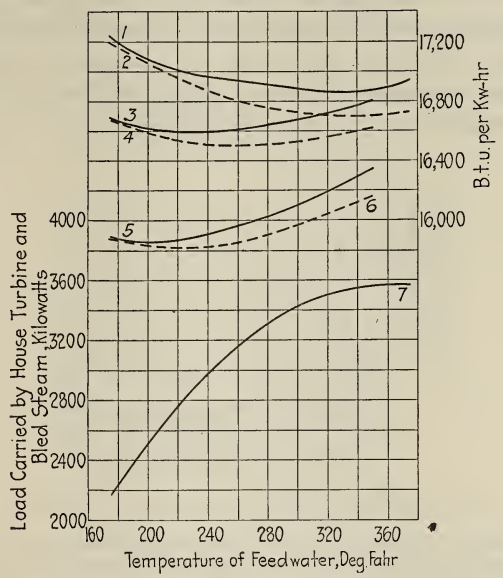

FIG. 17 FOUR-STAGE HEATING — CASE 2

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1 shows the heat consumption per kw-hr. for four-stage heating when not using economizers. Curve 2 is the same except that losses due to leakage from the low-pressure steam piping are not included. The difference between the two curves indicates the effect that the assumptions made as to these low-pressure steam losses have on the slope of the heat-consumption curves. Curves 3 and 5 are heat-consumption curves when using economizers of 18,180 and 47,000 sq. ft., respectively, and curves 4 and 6 are the same but do not include losses due to low-pressure steam leakage. Curve 7 shows the load carried by the steam used for heating the feedwater. Steam was derived from a 650-kw. house turbine and by bleeding the main unit. The temperatures for best economy are approximately 280 to 350 deg. when economizers are not used, 210 to 240 for the 18,180-sq.-ft. economizer arrangement and 190 to 220 deg. when using the 47,000-sq.-ft. economizer.

of best economy at 1400 kw. for Case 1 and 1500 kw. for Case 2, only the second-stage heater deriving steam by bleeding the main unit. The other arrangement used a house turbine with its point of best economy at 650 kw., additional steam required by the first-stage heater being obtained by bleeding the main unit. The feed-

water temperatures for best efficiency as indicated by curves 1 and 2 of Figs. 13 and 16 are seen to be approximately the same in either case, and about 25 deg. below that indicated by theoretical considerations alone. Both curves are rather flat over a range of 50 deg. in the vicinity of the point of best efficiency. The more efficient of the two arrangements, as with single-stage heating, is that one using the smaller house turbine, or the one bleeding the largest amount of steam from the main unit. This latter arrangement, with a feedwater temperature of 275 deg. fahr., showed a possible saving of approximately 320 B.t.u. per kw-hr. as compared with single-stage heating. The maximum efficiency using single-stage heating, however, was obtained with a feedwater temperature of approximately

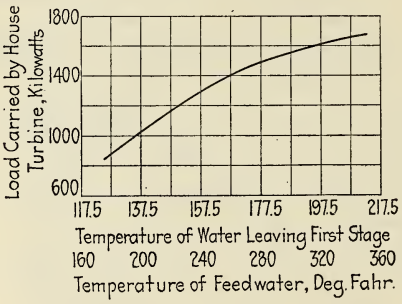

FIG. 18 DOUBLE-STAGE HEATING — CASE 2

(Load developed by house turbine used for double-stage heating when steam only for the second stage is bled from the main unit. The size of turbine selected was 1500 kw. and the load carried by it at the feedwater temperature for best economy was 1520 kw.)

200 deg. fahr. Comparing single-stage with double-stage heating on the basis that a temperature of 210 deg. fahr. was not to be exceeded, the difference between double-stage and single-stage heating is about 225 B.t.u. per kw-hr.

EFFECT OF ECONOMIZERS

12 Economizers in connection with single- and double-stage heating were applied to those stations which used a 650-kw. house turbine and bled the main unit, this arrangement being the more economical. The economizers were assumed as having a heat-recovery factor of 85 per cent and a heat-transfer rate of 5 B.t.u. per hour per deg. mean temperature difference between the flue gases and the water. Two sizes of economizers were applied to each

station to illustrate the effect of varying the size of the economizers. The boilers, as previously stated, were assumed to operate at approximately 225 per cent of their rated capacity, and their efficiency in this case was taken as 75 per cent, or 3 per cent less than that used when the stations had no economizers. The temperature of the flue gases entering the economizers in all cases was taken as 580 deg. fahr., which meant that for the type of boilers selected the percentage of rated capacity developed did not vary with the temperature of the feedwater. This, of course, required that the total

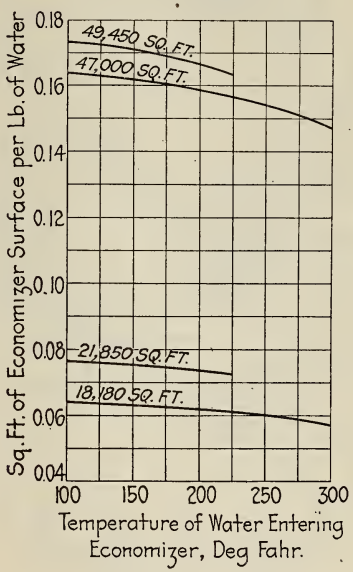

Fig. 19 Square Feet of Economizer Surface Used Expressed as a Ratio of the Amount of Feedwater Heated at Various Inlet Water Temperatures

(The 49,450-sq.-ft. economizer and the 21,850-sq.-ft. economizer were used for single-stage heating.)

operating capacity be slightly decreased as the temperature of the feedwater entering the boiler was increased. The weight of the flue gases per pound of coal burned was taken as 19 lb. and independently of the feedwater temperature, but, inasmuch as the weight of coal burned per pound of steam generated varied with the feedwater temperature, the weight of gas per pound of steam generated also varied.

13 Analyses of heat balances as affected by feedwater tem-

peratures did not involve considerations of boiler-room efficiency for those stations not using economizers. When economizers were used, however, the temperature of the flue gases leaving the economizer, for given equipment and operating conditions, was determined by the temperature of the water entering the economizer.

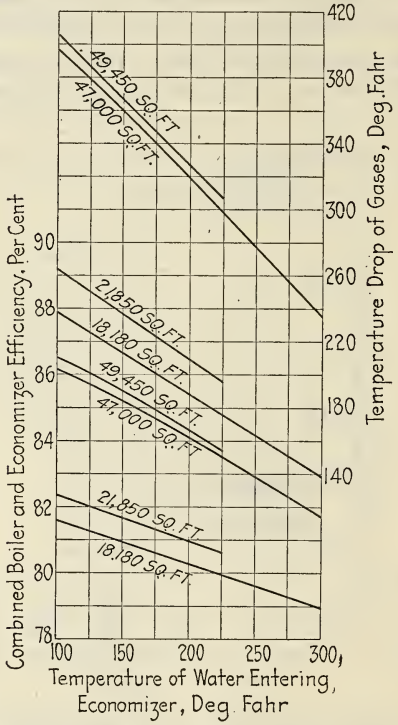

FIG. 20 DROP IN FLUE-GAS TEMPERATURE AND COMBINED BOILER AND ECONOMIZER EFFICIENCIES FOR VARIOUS ENTERING FEEDWATER TEMPERATURES

(The upper set of four curves show the drop in flue-gas temperature for various entering feedwater temperatures. The initial flue-gas temperature was 580 deg. fahr. The lower set of curves show the combined boiler and economizer efficiencies for various entering feedwater temperatures. The heat-recovery factor of the economizers was taken as 85 per cent and the boiler efficiency, without the economizers, as 75 per cent.)

Increasing the temperature of the water entering the economizer simultaneously increased the temperature of the flue gases so that the combined efficiency of the economizers and boilers was decreased. However, as the temperature of the feedwater leaving the heaters was increased above that of the condensate, the efficiency of converting steam to power, as previously demonstrated, increased,

thereby opposing the consequent decrease in efficiency of steam generation. The relative rates at which the efficiency of steam generation decreased and that of power generation increased determined the temperature of the feedwater for best efficiency, these rates being equal for the condition of best efficiency. The rate at which the efficiency of steam generation decreases with increase in

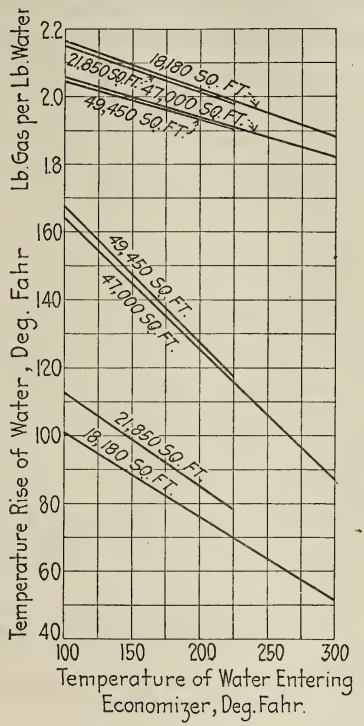

Fig. 21 Ratio of Weight of Flue Gases through Economizer to Weight of Water Flowing, and Rise in Temperature of Water

(The upper set of curves shows the ratio of the weight of flue gases passing through the economizers to the weight of water flowing. The data are based on 19 lb. of gas per pound of coal burned. The lower curves show the rise in the temperature of the water passing through the economizers for various entering water temperatures.)

feedwater temperatures depends on the relative area of the economizers and the boilers. The larger the economizer relative to the boiler, the more rapid is the rate at which this efficiency falls off. In consequence of this, the feedwater temperatures for best efficiency were found to be lower for the larger economizers than for the smaller ones, though the differences were not of great moment.

14 The temperature for best economy when economizers are used, as given by the heat-consumption curves 3 and 4 of Figs. 12 and 15 for single-stage heating, lies between 175 and 190 deg. fahr. for Case 1 and 165 and 175 deg. fahr. for Case 2. With double-stage heating, Figs. 13 and 16, the corresponding temperatures are 225 and 175 deg. fahr. for the smaller and larger economizer surfaces, respectively, of Case 2, and 230 and 200 deg. fahr. for Case 1. The temperatures for Case 1 are lower than those for Case 2, due

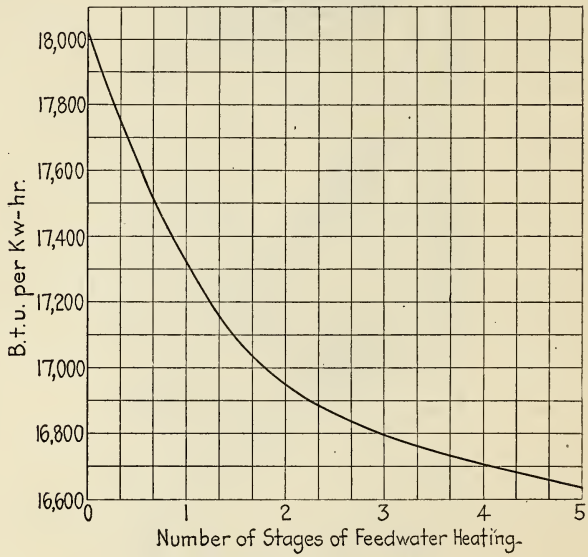

Fig. 22 Heat Consumption per Kw-hr. for Various Stages of Feedwater Heating — Case 2

(The curve illustrates that as the number of stages increases above two the value of the last stage decreases rather rapidly. The heat-consumption rates given by the curve do not include the losses due to the steam leakage from the low-pressure steam piping and are for the arrangements which do not use economizers.)

to the difference in the slopes of the Willans lines used. For each kilowatt-hour developed by the steam used for heating the feedwater in Case 1, the steam condensed in the main unit's condenser was reduced 12 lb., while for Case 2 this figure was 9 lb. The benefit derived from carrying load on steam used for heating the feedwater was therefore relatively less for Case 2 than for Case 1, while the economizer and boiler efficiency remained the same. The differences in temperature are small, however, and indicate that with fair

accuracy the desirable feedwater temperature may be considered as a range which is largely independent of the characteristics of the steam equipment used. The economizers used for double-stage heating were slightly smaller than those used for single-stage heating, commercial considerations indicating that this was justified. The difference in the sizes of the economizers did not, however, materially affect the feedwater temperature for best efficiency. The rate-of-heat-consumption curves are rather flat over a considerable range in proximity to the temperature for best economy, and increasing the area of the economizers even to the extent of doubling them need not require large changes in the temperature of the feedwater. The various data used in connection with the economizers are given in curves, Figs. 19, 20 and 21.

COMPARISON OF RESULTS WITH AND WITHOUT ECONOMIZERS

15 A comparison of feedwater temperatures for best economy as indicated by Case 1 and Case 2, respectively, shows that when economizers are not used ordinary variations in the Rankine-cycle efficiency of the bled steam, the efficiency curve for the house turbine, and the slope of the Willans line of the main unit have no considerable effect. When economizers are used, the temperatures are reduced by decreasing the slope of the Willans line of the main unit, but, as previously stated, the influence is not large. It is evident that for the purpose of establishing the proper feedwater temperature for best economy minute accuracy in the determination of the various turbine efficiencies is not required. The overall efficiency of the entire station is influenced, of course, by these efficiencies, but the best feedwater temperature changes only slightly with them. For this study a constant efficiency for bled steam was used. However, the house-turbine efficiencies were considered as being a function of the feedwater temperature, and a comparison of the temperatures for best economy obtained when using a house turbine alone with those obtained when using a smaller house turbine together with bleeding the main unit illustrates the influence of the variation in the Rankine-cycle efficiency. The difference in the temperatures that were obtained is between 10 and 15 deg. fahr., but as the curves are flat this is of no great consequence.

MULTIPLE-STAGE HEATING

16 By similar methods of computation, heat balances for four-stage heaters with and without the use of ceonomizers were determined and the results are presented by the curves of Fig. 17. The average Rankine-cycle efficiency for the bled steam in this case was taken at 79 per cent instead of 80 per cent, as the average efficiency over four stages would probably be somewhat less than that for one stage. A curve, Fig. 22, is also given showing the heat consumption per kilowatt-hour for the various methods of heating the feedwater worked out for Case 2, using the 650-kw. house turbine. As the number of heating effects increase, the value of the last effect decreases, which is to be expected. It is interesting to observe also that as the number of effects increase, the heat-consumption curves for the various stations flatten out in proximity to the temperature for best economy and that it would therefore seem undesirable to go beyond a certain feedwater temperature regardless of the number of stages. When economizers of 21,850 sq. ft. are used the gains in efficiency referred to a basis of no heating for single-, double-, triple-, and quadruple-stage heating are 1.86, 2.88, 3.35, and 3.64 per cent, respectively. A proper comparison of two-, three- and four-stage heaters would require a study of investment and operating charges, and the curves are given only as an indication of what may be expected from the viewpoint of the thermal efficiency when the number of heating effects are increased. As stated above, with increasing number of effects the value of the last heating decreases, and in determining the proper number of stages consideration must be given to the investment and also to the operating problems encountered.

THE FORMULAS USED

17 The various formulas and data used in making this study are given in the following pages. In general, when economizers are not used the temperature for best economy occurs in the neighborhood of that which gives a maximum for the power generated by the steam used for heating the feedwater. When economizers are used the temperature for best economy is given rather closely by determining the temperature which makes —

$$\frac{\Delta L}{\Delta t} = \frac{W'\left(\dfrac{1}{\dfrac{W'}{aT} + \dfrac{W'b}{0.475GHi} + 0.5}\right)}{n\,(H_S - t_1 + 32) - h}$$

where $\frac{\Delta L}{\Delta t}$ = rate of increase of load carried by steam used for heating the feedwater with increase of feedwater temperature

W' = weight of water passing through economizer per hour

G = weight of gases per pound of coal

i = percentage of recoverable heat

b = heat value of coal in B.t.u. per lb.

n = slope of Willans line of main unit

H_S = heat content of steam in initial state

a = total area of economizer in sq. ft.

T = rate of heat transmission in B.t.u. per hr. per sq. ft. per deg. mean temperature difference

h = B.t.u. required by the house turbine to produce 1 kw-hr.

t_1 = temperature of condensate from main unit

t = temperature of feedwater entering economizer

H = total over all heat requirements of station when feedwater is at temperature t, i. e., B.t.u. in coal fired.

$\Delta L/\Delta t$ may be found by determining the increments in L for definite increments in t and dividing the increments in L by the chosen increments of t, using for single-stage heating the formula —

$$L = \frac{rBS'(t - t_1)}{3415C + rBn(t - t_1)}$$

where r = Rankine-cycle efficiency of house turbines

B = adiabatic heat drop from initial steam condition to the temperature of the feedwater

S' = weight of steam consumed by the main unit when carrying the total load including that of all auxiliaries

C = heat available per lb. of heating steam for raising temperature of feedwater.

The value of

$$\frac{W'\left(\dfrac{1}{\dfrac{W'}{aT} + \dfrac{W'b}{0.475GHi} + 0.5}\right)}{n(H_S - t_1 + 32) - h}$$

does not vary greatly with the feedwater temperature, so that one determination suffices. As the value of this expression decreases, the feedwater temperature for the best economy increases. It is of value, therefore, in judging the influence of various factors, though for determining the proper feedwater temperature the formulas in the Appendix should be used.

AIR ECONOMIZERS

18 If air for the boilers is preheated by means of air economizers and no feedwater economizers are used, such preheating does not affect the feedwater temperature for best economy. However, the air may be preheated by exhaust steam similarly to heating feedwater. Also air economizers and exhaust-steam air heating may be used with or without the coincident use of water economizers, and with single- or multiple-stage feedwater heating. If air is heated by exhaust steam alone, the problem of determining its temperature for maximum efficiency is not different from that for determining the temperature of feedwater heated only by exhaust steam. This, however, is not the case if both air and water economizers are used simultaneously.

19 Air and water economizers, if used simultaneously, may be placed either in parallel or in series. If placed in parallel the effect of the air economizer is to reduce the amount of gases available for heating the feedwater, and so alter the operating characteristics of the water economizers so far as these are determined by the ratio of the waste gases to the water heated. If the air heater is placed in series with the water economizer and in the coldest part of the flue gases, the leaving temperature of these gases is no longer determined by the temperature of the boiler feedwater. For such an arrangement it would appear desirable to heat the boiler feedwater to that feedwater temperature giving the maximum efficiency for the conversion of steam energy to electrical power. The feedwater would then be further heated in a water economizer while the temperature of the flue gases leaving the water economizer could be reduced to a desirable point, justified, of course, by the efficiency of the air economizers and their cost. On this basis the temperature of the feedwater entering the economizers would correspond to that previously determined when no economizers were used. The air economizers in this case, of course, would have to reduce the temperature of the flue gases below that established for the use of water economizers alone.

CONCLUSIONS

20 So many factors enter into the determination of the proper feedwater temperature of a plant that figures determined for one station should not be directly applied to another. The tem-

perature should be lower for plants using single-stage heating than for those using multiple-stage heating, and also should be lower for plants using economizers than for plants not using economizers. Efficiencies of auxiliary apparatus as well as of the main generating units have their influence. If the feedwater temperature be raised much above 212 deg. fahr., factors incident to the use of pressures well above atmosphere in the auxiliary exhaust piping come into play. The proper feedwater temperature is dependent somewhat upon the initial steam pressure and temperature and on the temperature of the condensate, increasing as these increase. It is quite impossible, therefore, to determine, except within wide limits, feedwater temperatures applicable to all plants, and the purpose has been rather to indicate in a broad way the effect of feedwater temperatures on power-plant efficiency, leaving out the matter of costs, and to give some basis for estimating the sacrifice in fuel made to assure practicable operation of the feedwater-heating system as laid out for a particular station.

APPENDIX NO. 1

DATA

21 The following data were used as bases for the heat-balance study o this paper:

Net station load.....25,000 kw.
Load on auxiliary bus.....1,300 kw.
Gross station load.....26,300 kw.
Steam consumption of the main unit when carrying the gross station load:
Case 1: 279,000 lb. per hr.
Case 2: 270,000 lb. per hr.
High-pressure drips.....1000 lb. per hr.
Condensate losses.....1000 lb. per hr.
High-pressure steam losses.....3000 lb. per hr.
Radiation losses from low-pressure steam:
2 per cent of total heat in steam used for heating the feedwater
Pressure of steam at throttle.....330 lb. per sq. in. gage
Boiler pressure.....350 lb. per sq. in. gage
Superheat.....200 deg. fahr.
Heat content of boiler steam.....1326 B.t.u. per lb.
Heat content of steam at throttle.....1324 B.t.u. per lb.
Vacuum on main unit.....29 in. Hg.
Temperature of condensate.....75 deg. fahr.
Temperature of make-up water entering evaporator.....60 deg. fahr.
Slope of Willans line of main unit.....Case 1: 12 lb. per kw-hr.
Case 2: 9 lb. per kw-hr.
Heat content of the high-pressure drips recovered.....390 B.t.u. per lb.
Radiation, friction and generator losses of the house turbine in kw.:
650-kw. turbine, 65 kw.
1500-kw. turbine, 135 kw.
Radiation losses from bled steam on passing through the main unit:
1 per cent of load developed by the bled steam
Internal Rankine-cycle efficiency of bled steam based on
steam pressure after throttle.....Case 1: 67 per cent
Case 2: 80 per cent
Boiler efficiency when not using economizers.....78 per cent
Boiler efficiency when using economizers but not including
the economizer efficiency.....75 per cent
Coefficient of heat transmission through economizers.....5 B.t.u. per sq. ft.
per deg. mean temperature difference between flue gases and water
Cost of economizers per sq. ft. of surface.....$4.00
Cost of boilers per sq. ft. of surface.....$4.50
Annual charges against economizers which vary as their size:
20 per cent of their cost
Annual charges against boilers.....15 per cent of their cost
Use factor of economizers.....50 per cent and 100 per cent
Use factor of boilers.....85 per cent

Temperature of gases entering economizers.....580 deg. fahr.
Specific heat of flue gases.....0.2375
Percentage of recoverable heat recovered by economizers.....85 per cent
Flue gases per lb. of coal.....19 lb.
Heating value of coal.....13,500 B.t.u. per lb.
Cost of fuel.....$5.00 per ton
Ratio of the load developed by the boilers to the full-load rating of the boilers when using economizers.....2.25

APPENDIX NO. 2

FORMULAS

I — Formula for determining the load developed by the steam used in a single-stage heater when the steam is obtained from a house turbine and by bleeding the main unit

22 *Nomenclature:*

M = make-up water in lb. per hr.
C = condensate from the main unit, after deducting losses, in lb. per hr.
S_B = steam bled from the main unit in lb. per hr.
S_H = steam derived from the house turbine in lb. per hr.
D = high-pressure drips in lb. per hr.
R = radiation losses in per cent of the heat in S_B and S_H
L_H = load developed by the house turbine in kw.
L_B = load developed by the steam bled from the main unit in kw.
W_H = water rate of the house turbine in lb. of steam per kw-hr.
W_B = pounds of steam bled per kw-hr. developed by the bled steam
C_1 = steam supplied to the main unit when carrying the full station load, including that of all auxiliaries when no steam is bled
a = condensate losses
$C_1 - a$ = condensate to be heated when the main unit carries the entire station load and no steam is bled
T = temperature of the feedwater
L = total load developed by the steam subsequently used for heating the feedwater
h_m = heat content of make-up water in B.t.u. per lb.
h_c = heat content of condensate in B.t.u. per lb.
H = heat content of high-pressure steam in B.t.u. per lb.
h_d = heat content of high-pressure drips in B.t.u. per lb.
n = slope of Willans line of main unit in the region of the load carried by it
l_H = radiation, friction, and generator losses of the house turbine in kw.
l_B = radiation losses of the bled steam passing through the main unit in per cent of the load developed by the bled steam.

23 Since the heat content of the mixture leaving the heater is equivalent to the heat content of the fluids entering the heater,

$$(S_B+M+C+S_H+D)(T-32)=h_mM+h_cC+(1-R)[HS_H-(L_H+l_H)3415] + (1-R)[HS_B-L_B(1+l_B)3415]+Dh_d$$

$$C = C_1 - a - nL$$

$$S_H = L_HW_H$$

$$L_B = L - L_H \text{ and}$$

$$S_B = (L - L_H)W_B$$

from which —

$$L=\frac{h_mM+h_c(C_1-a)+(1-R)HL_HW_H-(1-R)(L_H+l_H)3415-(1-R)HL_HW_B+(T-32)(W_B-n)+h_cn-(1+l_B)3415L_H(1-R)+Dh_d-(T-32)(M+C_1-a+L_HW_H+D-L_HW_B)}{(1-R)HW_B+(1-R)(1+l_B)3415}$$

II — Formula for determining the load carried by the steam used in a single-stage heater when the steam is derived only from a house turbine

24 *Nomenclature:* same as for Formula I, except for the following:

S_1 = steam consumption of the house turbine at full load

S_2 = steam consumption of the house turbine at half-load

K = full-load rating of the house turbine

$$S_H = \frac{2(S_1 - S_2)L}{K} + 2S_2 - S_1$$

W_1 = full-load water rate in lb. of steam per hr.

W_2 = half-load water rate in lb. of steam per hr.

$$S_H = L(2W_1 - W_2) + K(W_2 - W_1)$$

25 Since the heat content of the mixture leaving the heater equals that of the fluids entering the heater,

$$[(M + C_1 - a - nL + L(2W_1 - W_2) + K(W_2 - W_1) + D)](T - 32) = h_mM + h_cC + (1 - R)HS_H - (1 - R)(L + l_H)3415 + Dh_d$$

from which —

$$L = \frac{(T-32)[(M+C_1-a+K(W_2-W_1)+D)]-h_mM-h_c(C_1-a)-(T-32)n-(T-32)(2W_1-W_2)-h_cn+(1-R)HK(W_2-W_1)+(1-R)l_H3415-Dh_d}{(1-R)H(2W_1-W_2)-3415(1-R)}$$

III — Formula used for determining the load developed by steam used in a double-stage heater, the first stage drawing steam from a house turbine and the second stage obtaining steam by bleeding the main unit

26 Since the heat given up by the steam equals the increased heat content of the water heated in the heater, using the previous nomenclature, and using T_1 and T_2 to designate the temperatures of the feedwater entering the first- and second-stage heaters, respectively, and T_3 for the temperature leaving the second-stage heater.

$$(1 - R)[HL_HW_H - (L_H + l_H)3415] - L_HW_H(T_2 - 32) = (M + C_1 - a - nL + D)(T_2 - T_1)$$

for the first-stage, and —

$$(1 - R)\,[HL_BW_B - L_B\,(1 + l_B)\,3415] - L_BW_B\,(T_3 - 32)$$
$$= (M + C_1 - a - nL + D + L_HW_H)\,(T_3 - T_2)$$

for the second-stage heater.

$$W_H = (2\,W_1 - W_2) + \frac{K}{L_H}(W_2 - W_1)$$

from which —

$$L_H = \frac{(M + C_1 - a - nL + D)\,(T_2 - T_1) + (1 - R)\,3415l_H - (2W_1 - W_2)\,[H\,(1 - R) - T_2 + 32] - \dfrac{K\,(W_2 - W_1)\,[H\,(1 - R) - T_2 + 32]}{(1 - R)\,3415}}{}$$

$$L_B = \frac{(M + C_1 - a - nL + D + L_HW_H)\,(T_3 - T_2)}{(1 - R)\,[HW_B - (1 + l_B)\,3415] - W_B(T_3 - 32)}$$

$$L = L_H + \frac{[M + C_1 - a + D + K\,(W_2 - W_1) + (2W_1 - W_2 - n)\,L_H]\,(T_3 - T_2)}{(1 - R)\,[HW_B - (1 + l_B)\,3415] - W_B\,(T_3 - 32) + n\,(T_3 - T_2)}$$

Substituting the value of L in the equation for L_H permits solving for L_H.

IV — FORMULA FOR DETERMINING THE LOAD DEVELOPED BY STEAM USED IN HEATING WHEN THE STEAM IS DERIVED FROM A 650-KW. HOUSE TURBINE AND BY BLEEDING THE MAIN UNIT AT TWO, THREE OR FOUR POINTS ACCORDINGLY AS TWO-, THREE- OR FOUR-STAGE HEATERS ARE USED

27 The notations used are the same as previously used except for the following:

L'_B, L''_B, L'''_B and L''''_B designate the loads developed by the bled steam used in the first-, second-, third- and fourth-stage heaters, respectively

W'_B, W''_B, W'''_B and W''''_B designate the water rates for that part of the load carried by steam bled for the first-, second-, third- and fourth-stage heaters, respectively.

28 Since the heat given up by the steam entering the heater equals the increased heat content of the water heated,

$$(1 - R)\,[HL_HW_H - (L_H + l_H)\,3415] - L_HW_H\,(T_2 - 32)$$
$$+ (1 - R)\,[HL'_BW'_B - L'_B\,(1 + l_B)\,3415] - L'_BW'_B\,(T_2 - 32)$$
$$= (M + C_1 - a - nL + D)\,(T_2 - T_1)$$

for the first-stage heater, from which —

$$L_B' = \frac{(T_2 - T_1)\,(M + C_1 - a - nL + D) - L_HW_H\,[(1 - R)\,H - T_2 + 32] + W_B'\,[(1 - R)\,H - T_2 + 32] - \dfrac{(1 - R)\,3415\,(L_H + l_H)}{(1 - R)\,(1 + l_B)\,3415}}{}$$

$$L_B'' = \frac{(M + C_1 - a - nL + D + L_HW_H + L_B'W_B')\,(T_3 - T_2)}{(1 - R)\,[HW_B'' - (1 + l_B)\,3415] - W_B''\,(T_3 - 32)}$$

$$L_B''' = \frac{(M + C_1 - a - nL + D + L_HW_H + L_B'W_B' + L_B''W_B'')\,(T_4 - T_3)}{(1 - R)\,[HW_B''' - (1 + l_B)\,3415] - W_B'''\,(T_4 - 32)}$$

$$L_B'''' = \frac{(M+C_1-a-nL+D+L_HW_H+L_B'W_B'+L_B''W_B''+L_B'''W_B'')\,(T_5-T_4)}{(1-R)\,[HW_B''''-(1+l_B)3415]-W_B''''\,(T_5-32)}$$

$$H_H = (1-R)\,H - T_2 + 32 - \frac{(1-R)}{W_H}\left(1+\frac{l_H}{L_H}\right)3415$$

$$H_B' = (1-R)\,H - T_2 + 32 - \frac{(1-R)}{W_B'}\,(1+l_B)\,3415$$

$$H_B'' = (1-R)\,H - T_3 + 32 - \frac{(1-R)}{W_B''}\,(1+l_B)\,3415$$

$$H_B''' = (1-R)\,H - T_4 + 32 - \frac{(1-R)}{W_B'''}\,(1+l_B)\,3415$$

$$H_B'''' = (1-R)\,H - T_5 + 32 - \frac{(1-R)}{W_B''''}\,(1+l_B)\,3415$$

H_H, H_B', H_B'', H_B''' and H_B'''' represent the heat content, not including that of the liquid, in the exhaust steam from the house turbine and in the steam bled for the first, second, third and fourth stage heaters respectively.

$$\text{Let}\left[\frac{T_3-T_2}{H_B''Wx''}+\left(\frac{T_3-T_2}{H_B''}+1\right)\frac{(T_4-T_3)}{H_B'''W_B'''}\right.$$
$$\left.+\left(\frac{T_3-T_2}{H_B''}+1\right)\left(\frac{T_4-T_3}{H_B'''}+1\right)\left(\frac{T_5-T_4}{H_B''''W_B''''}\right)\right]=A$$

$$\text{Then } L = L_B'\left[W_B'\left(1+\frac{H_B'}{T_2-T_1}\right)A+1\right]+L_H\left[W_H\left(1+\frac{H_H}{T_2-T_1}\right)A+1\right]$$

$$L'_B = \frac{(T_2-T_1)\left[M+C_1-a-nL_H[AW_H\left(1+\frac{H_H}{T_2-T_1}\right)+1]+D\right]-L_HW_HH_H}{W_B'H_B'+n\left[AW_B'\left(1+\frac{H_B'}{T_2-T_1}\right)+1\right](T_2-T_1)}$$

$$L_B'' = \frac{T_3-T_2}{H_B''W_B''}\left[L_B'W_B'\left(1+\frac{H_B'}{T_2-T_1}\right)+L_HW_H\left(1+\frac{H_H}{T_2-T_1}\right)\right]$$

$$L_B''' = L_B''W_B''\left(1+\frac{H_B''}{T_3-T_2}\right)\left(\frac{T_4-T_3}{H_B'''W_B'''}\right)$$

$$L_B'''' = L_B'''W_B'''\left(1+\frac{H_B'''}{(T_4-T_3)}\right)\left(\frac{T_5-T_4}{H_B''''W_B''''}\right)$$

The load on the house turbine L_H is determined by the capacity of the house turbine which in the case considered was 650 kw.

V — Formula for determining the steam passing through to the condenser of the main unit

$C_1 - nL$ = steam condensed

VI — Formula used to determine the combined boiler and economizer efficiencies when using economizers, and the size of the economizer for best commercial efficiency

29 *Nomenclature:*

g = weight of flue gas per lb. of coal
S = sq. ft. of economizer surface per lb. of water

H_S = heat content of steam as delivered by the boilers
T_2 = temperature of the water leaving the economizers
T_1 = temperature of the water entering the economizers
r = charges against economizers in percent of their initial cost (These charges are those which vary with the size of the economizer)
C = cost of economizers in dollars per sq. ft.
r' = fixed charges against boilers in per cent of their initial cost
a = use factor for economizers
b = use factor for boilers
T_{g1} = initial temperature of the flue gases
T_{w1} = initial temperature of the water
$D = T_{g1} - T_{w1}$
T = coefficient of heat transmission through economizers
G = pounds of flue gas per lb. of water
R = rise in temperature of the feedwater
e = boiler efficiency not including economizer
F = cost of fuel in dollars per ton
L = ratio of load developed by boilers to the full-load rating of the boilers
C_B = cost of boilers in dollars per sq. ft.
B = heat value of coal in B.t.u. per lb.
i = percentage of recoverable heat.

30 The areas of the economizers used were determined by the following formula:

$$S = \frac{K\sqrt{D} - \frac{1}{T}}{\frac{1}{0.475iG} + 0.5} \qquad K = \sqrt{\frac{a}{TrC}\left(\frac{4.375F}{Be} + \frac{10C_Br'}{33400Lb}\right)}$$

S represents the area of economizer surface per pound of water flowing through the economizer. The area of the economizer so determined gives the best commercial efficiency, assuming that the cost of the economizers, and the variable charges against them which vary with their size, may be expressed as a fixed ratio of their area.

31 The increased efficiency of steam generation due to the addition of economizers was determined by the following formula:

$$e = \frac{100\,g\,R}{G\,B}$$

in which —

R = rise in temperature of water passing through the economizer

$$= \frac{D}{\frac{1}{ST} + \frac{1}{0.475\,Gi} + 0.5}$$

$$G = \frac{g}{Be}\left(H_S + 32 - T_1 - \frac{D}{\frac{1}{ST} + \frac{1}{0.475\,G\,i} + 0.5}\right)$$

This equation is readily solved by first approximating G.

DISCUSSION

G. G. Bell.[1] Mr. Helander has presented a very interesting paper on the effect of feedwater heating on plant economy. He has made a very complete study of the subject, and shows the theoretical relation between plants equipped with large and small house turbines, both with high boilers without economizers and medium-sized boilers with economizers.

In the spring of 1922, two more 30,000-kw. units were purchased for the Windsor station, the boiler drum pressure was raised from 250 to 350 lb., and on account of this higher pressure, steel-tube economizers were substituted for the cast-iron economizer which had been installed with the first 16 boilers in the plant.

The troubles which other plants had had with corrosion in steel-tube economizers necessitated the installation of deaërating apparatus. Manufacturers of deaërating apparatus claimed that air separation was easier at temperatures above 160 deg. fahr. Experience with a closed system of feedwater heating which prevented enrichment indicated that an average oxygen content in the feedwater of $\frac{1}{4}$ c.c. per liter could be maintained, provided the feedwater temperature was held at 210 deg. fahr. Investigations of an existing plant equipped with economizers had indicated some advantages in increasing the feedwater temperature to 210 deg. fahr. by utilizing the exhaust steam from a house turbine of sufficient size to supply the auxiliaries with power; although subsequent investigations have demonstrated that probably there would be a slightly higher saving if the study had been made for a temperature of 190 instead of 210 deg.

In the new addition it was decided to heat the condensate by steam bled from the main unit instead of exhaust steam from the house turbine, and it was thought that the best temperature at which the feedwater should enter the economizers should not be less than 210 deg. on account of the more efficient use of the steam in the main unit. To check this a study was made, as a result of which it was decided to heat the feedwater to the highest temperature possible by extracting steam from the thirteenth stage.

Upon the preparation of Mr. Helander's paper, these studies were revised. Additional data on the turbine was obtained so as to give complete information for bleeding all stages from the eighth

[1] West Penn Power Co., Pittsburgh, Pa.

to the fifteenth inclusive, and the effect on the economy of the station of single-, double- and triple-stage bleeding within these limits was studied.

Data on steam extracted from the 30,000-kw. G. E. Curtis turbine at a load of 28,000 kw., steam conditions at throttle 300 lb. and 200 deg. and 1 in. back pressure, are shown on Fig. 23.

Curve No. 1 shows the absolute pressure at the various stages.

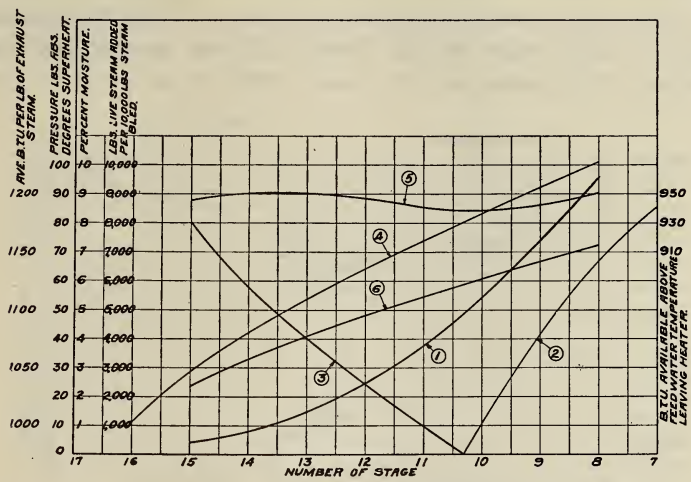

FIG. 23 DATA ON STEAM EXTRACTED FROM 30,000-KW. G. E. CURTIS TURBINE

Steam condition 300 lb. 200 deg. 1 in. load on turbine 28,000 kw.

No. 1 Absolute pressure at various stages

No. 2 Deg. superheat in bled steam

No. 3 Per cent moisture in bled steam

No. 4 Average B.t.u. per lb. of bled steam

No. 5 B.t.u. available for heating feedwater to maximum practical temperature by bled steam

No. 6 No. of lb. of live steam added per 10,000 lb. of steam bled.

Curve No. 2 shows the degree of superheat in the steam extracted from the various stages.

Curve No. 3 shows the percentage of moisture in the steam extracted from the various stages.

Curve No. 4 shows the average B.t.u. in each pound of steam bled.

Curve No. 5 shows the number of B.t.u. available for heating the feedwater to the maximum possible temperature with steam extracted from any stage.

Curve No. 6 shows the number of pounds of live steam that

must be added at the throttle in order to permit of the extraction of 10,000 pounds of steam at various bleeding points.

In determining the temperature to which the feedwater could be heated by each of the various stages, it was assumed that there was a loss between the turbine and the heater of 1 pound when steam was extracted from the fifteenth stage, of $1\frac{1}{2}$ pounds when steam was extracted from the fourteenth stage, and 2 pounds when steam was extracted from the thirteenth or any higher stage; and that in all cases the maximum temperature to which the feedwater could be heated was 10 deg. lower than the temperature corresponding to the pressure of the steam at the heater.

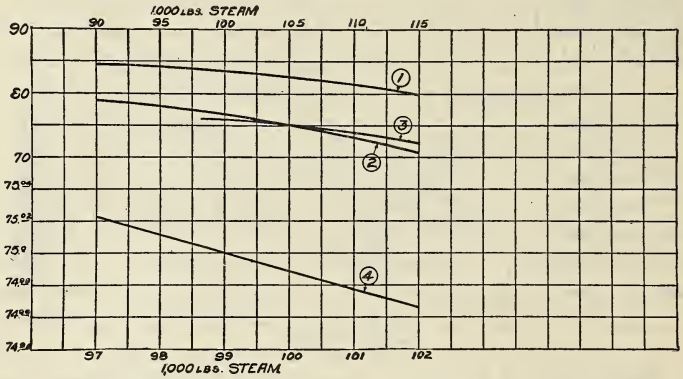

Fig. 24 Efficiency of 14 High, Cross Drum B. & W. Boiler

8341 sq. ft. of steel tube economizer. Steam pressure at drum 325 lb. Temperature at superheater outlet 225 deg. fahr. Feedwater temperature 210 deg. fahr.

The boilers purchased for Windsor are 14 tubes high and 42 tubes wide, Babcock & Wilcox cross-drum boilers, the boilers being equipped with a slag screen and Babcock & Wilcox inclined baffle. The front headers are set 21 feet above the floor, the drum center being 35 ft. 3 in. Four boilers are provided per unit, although it was assumed that when the turbine was operating at its point of best efficiency, which is 28,000-kw., three boilers would supply steam for the unit, the output of each boiler being about 100,000 lb. when feedwater is supplied at a temperature of 210 deg.

The extra boiler capacity would be utilized in reducing the rating on the boilers in the older section of the plant which are not as efficient or as liberally stokered as the newer boilers.

Fig. 24 shows the efficiency of the new boiler and steel-tube economizer at various ratings.

Curve No. 1 shows the combined efficiency of boiler and economizer.

Curve No. 2 shows the efficiency of the boiler alone.

Curve No. 3 shows the boiler efficiency corrected for the effect of change in capacity on the efficiency of the economizer.

Curve No. 4 is that portion of the preceding curve which is used, amplified so as to permit of reading any slight variation in the relative efficiency of the boiler when operating at sight differences in output.

These figures are based upon 12 per cent CO_2. It is necessary in order to compare the results of bleeding from various

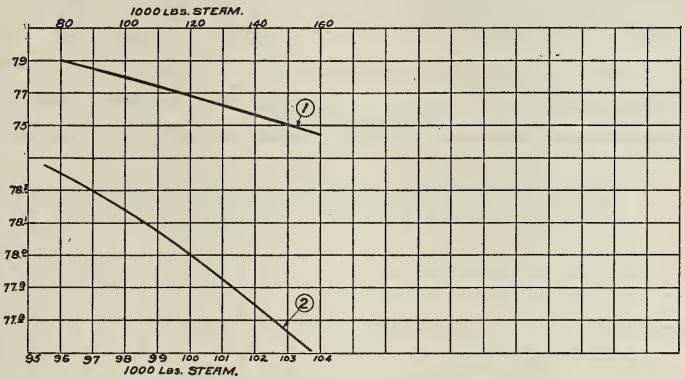

FIG. 25 EFFICIENCY OF 20 HIGH, CROSS DRUM B. & W. BOILER

No economizer. Steam pressure at drum 325 lb. Temperature at superheater outlet 225 deg. Feedwater temperature 210 deg.

stages to work to a fraction of a per cent. All computations were checked by the comptometer.

Fig. 25 gives similar data for a 20-tube high boiler without economizers.

Curve No. 1 gives the efficiency of the boiler at various ratings.

Curve No. 2 is that portion of the preceding curve which is used, amplified so as to permit of reading any slight variation in the relative efficiency of the boiler when operating at slight differences in output.

In connection with the 14-tube high boiler, 8341-sq. ft. steel economizers are installed. The following tabulation gives the rise in the economizer when operating at a constant output of 100,000 lb. of steam per hour with feedwater temperatures of 135, 170, 210 and 250 deg. The figures are derived from guarantees based

on 210-deg. feedwater and are proportional to the arithmetical mean of the temperature differences.

	Feedwater temperatures, deg. fahr			
	135	170	210	250
Gas temperature entering economizer, deg. fahr.	595	595	595	595
Gas temperature out of economizer, deg. fahr.	297	319	345	370
Drop in economizers, deg. fahr.	298	276	250	225
Average gas temperature, deg. fahr.	446	457	470	482.5
Water inlet to economizer, deg. fahr.	135	170	210	250
Estimated rise in economizer, deg. fahr.	128	118	107	96
Temperature water leaving economizer, deg. fahr.	263	288	317	346
Average water temperature, deg. fahr.	199	229	263.5	298
Average thermal difference in economizer, deg. fahr.	247	228	206.5	184.5
Rise in economizers, deg. fahr.	128	118.4	107	95.8

These results are plotted on Fig. 26, which also shows a comparison of single-extraction heating and heating the condensate by

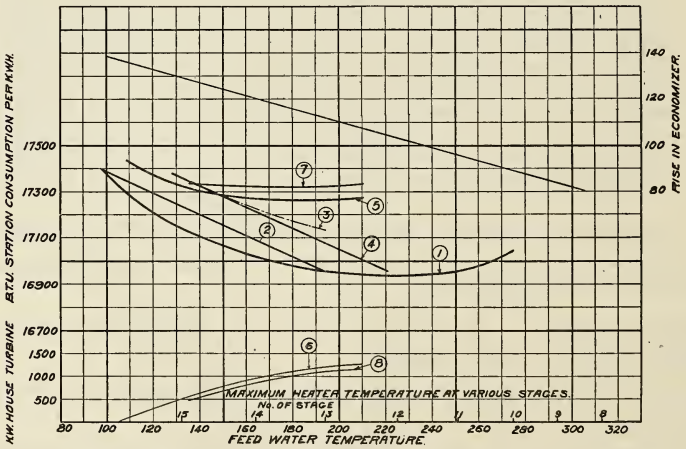

Fig. 26 Comparison of Effect on Station Economy of Heating Feedwater by Exhaust Steam from House Turbine, Efficient Duplex Driven Auxiliaries and Single-Stage Extraction Heating

North extension of Windsor power station. 30,000-kw. G. E. turbines. 28,000-kw. load. 14 high, cross drum B. &. W. boilers, ind. draft. 60 per cent economizer. Steam at throttle 300 lb., 200 deg., 1 in.

exhaust steam from a house turbine if the Windsor plant is equipped with a 14-tube high boiler and 8341-sq. ft. economizer.

Curve No. 1 shows the heat consumption of the plant if the

turbine is arranged for single-stage bleeding. These results are arrived at by calculating the heat requirements with bleeding at various stages from the tenth to the fifteenth, and drawing a curve through the points thus obtained. The curve represents the results which might be obtained if the turbine were designed with an infinite number of stages and could be bled at any one of them. As a matter of practice these results can only be obtained when steam is bled from any one of the six stages coming within the limit of this curve, and the feedwater is heated as hot as possible by the steam extracted from the turbine.

Curve No. 2 shows the effect of bleeding from the thirteenth stage, and throttling the amount of steam bled so as to get a varying temperature. This is practically a straight line. For the purpose of this study the condensate was assumed to be heated first approximately 20 deg., by the heat contained in the steam escaping from the steam seal. Twenty degrees is also the amount the condensate would be heated if before passing into the first bleeder heater it first cooled the generator air and then absorbed the heat in the bearing and transformer oil for the unit.

On curves Nos. 1 and 2 it was assumed that the boiler feed pump was motor-driven. If in place of a motor-driven boiler feed pump a steam-driven pump is used, Curve No. 3 would represent the heat requirements of the station. When sufficient steam is bled from the main unit to heat the feedwater to the maximum obtainable from the thirteenth stage, the heat consumption in the plant equipped with a motor-driven pump is approximately 1 per cent less than in a plant equipped with steam-turbine-driven pump. This is when the exhaust steam from the steam-driven boiler feed pump is discharged into the same heater as the steam bled from the main unit. When using a steam-driven boiler feed pump the minimum temperature is increased from 97.5 to 127.5 deg fahr.

Curve No. 4 shows the results if steam from the boiler feed pump is discharged to a separate heater and used to heat the feedwater sufficiently above the temperature of the feedwater leaving the extraction heater to get the necessary reëvaporation to permit the deaërator to function satisfactorily. Where the pressure will permit, the exhaust steam from the turbine glands is discharged into the same condenser as the exhaust steam from the boiler feed pump. The curve indicates that approximately the same results can be obtained by using the turbine-driven pump as the motor-driven pump, provided that separate heaters are used

and the feedwater is heated approximately 25 deg. above that used with a motor-driven boiler feed pump.

Curve No. 5 shows the results which would be obtained if the feedwater were heated by exhaust steam from a house turbine. Guarantees were obtained on house turbines designed for three different back pressures, and in figuring this curve the steam consumption used in each case would apply only if a special house turbine were designed for the operating conditions under consideration. This curve indicates that there is very little difference in the economy of using a house turbine in connection with the 14-tube high boiler and 60 per cent economizer, with feedwater temperatures between 170 and 190 deg. Assuming that this house turbine had been bought for a back pressure corresponding to 180 deg., if the feedwater temperature were then varied by increasing or decreasing the output from the house turbine, the heat requirements of the plant over the range would be higher than shown in No. 5, and would only coincide with Curve No. 5 when using a feedwater temperature of 180 deg., the point for which this particular turbine was designed.

Curve No. 6 shows the amount of power which can be obtained from the house turbines when heating the feedwater to temperatures shown in Curve No. 5.

Curve No. 7 shows the results which would be obtained if the feedwater were heated by exhaust steam from efficient high-speed geared turbines driving the auxiliaries. In this case one design of turbine is used and operated with various back pressures, so that full advantage is not taken of the increase in vacuum which it is possible to get with the lower feedwater temperature. The water rate for various feedwater temperatures is as follows:

Final feedwater temperature, deg. fahr.	Back pressure on turbine exhaust, lb. per sq. in.	Water rate, lb. per. brake hp-hr.
135	4.53	16.5
170	7.99	17.0
210	16.13	21.0

The curve indicates that within the range of the temperatures under consideration there is practically no difference in the heat requirements of the plant, although if the small geared sets were so designed as to show an increase in economy with a decrease in

back pressure, it would pay to lower the feedwater temperature to the lowest point that was practicable and still prevent the sweating of the economizer tubes.

Curve No. 8 shows the amount of power generated by the geared turbine-driven sets. The reason that Curves Nos. 5 and 7 approach each other at the lower temperatures is that the house turbine is of larger capacity, and at low feedwater temperatures is operated at part loads, whereas in Curve No. 7 it is assumed that only enough of the geared driven units are driven to give the required temperature when each turbine is carrying the maximum load.

Fig. 27 shows a comparison of results on the Windsor plant if

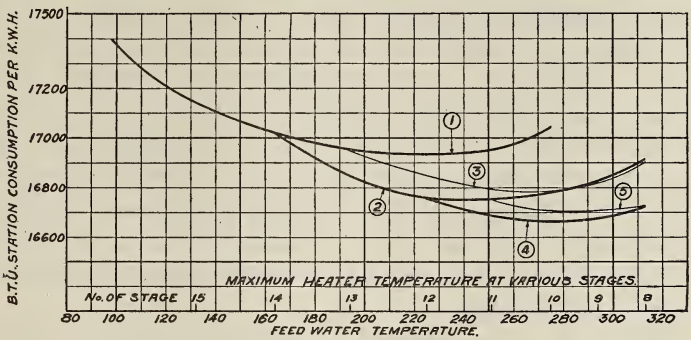

Fig. 27 Effect of Single-, Double- and Triple-Stage Extraction Heating on Feedwater Temperature and Station Economy

North extension of Windsor power station. 30,000-kw. G. E. turbines. 28,000-kw. load. Steam at throttle 300 lb., 200 deg., 1 in. 14 high, cross drum B. &. W. boilers. 60 per cent economizer, induced draft.

the turbine were arranged for single-, double- or triple-stage extraction heating if the plant were equipped with 14-tube high boilers and 8341-sq. ft. economizers.

Curve No. 1 shows the heat consumption of a plant arranged for single-stage bleeding at any of the temperatures within the limits of the curve. This curve is the same as Curve No. 1 of Fig. 26 and shows best results at a feedwater temperature of 225 deg. fahr. with a very slight increase in heat requirements by increasing or decreasing the feedwater temperature 25 or 30 deg.

Curve No. 2 shows the heat consumption of a plant arranged for bleeding from the fourteenth and a higher stage.

Curve No. 3 shows the heat consumption when bleeding from the thirteenth stage and a higher stage.

Curve No. 2 shows that minimum heat requirements are obtained by a combination of the fourteenth and twelfth or eleventh stages at a temperature of 225 or 251 deg. fahr.

Curve No. 4 shows that the best results with triple-stage heating are obtained with a combination of the fourteenth, twelfth and tenth stages, the best results being obtained at a temperature of 275 deg. fahr.

Curve No. 5 shows the heat consumption with triple-stage heating using the fourteenth, eleventh and a higher stage. This combination is not as efficient as that shown in Curve No. 4.

These studies are all made for a plant operated with a motor-driven boiler feed pump. The study indicates that there is very little difference between bleeding the eleventh, twelfth or the thirteenth stage with single-stage heating; and that for double-stage heating a combination of the fourteenth and twelfth or eleventh stages gives the best result, the heat requirements per net kw-hr. being about 16,750 as against 16,940 B.t.u. for single-stage heating. In triple-stage heating a combination of the fourteenth and twelfth and the tenth stages gives the best results, the heat requirements for triple-stage bleeding being about 16,660 B.t.u. as compared with 16,750 B.t.u. for the double-stage heating, and 16,940 B.t.u. for single-stage heating.

While there are a number of points to consider in obtaining plant heat requirements, the work can be reduced to a comparatively simple form; and with a set of curves as are given in Fig. 23 giving information for various stages and temperatures, etc., a point can be determined every twenty minutes by using a slide rule. However, a slide rule is not accurate enough to give smooth curves.

In order to illustrate the method used in making these computations, the formulas used are given below:

A = the number of pounds of steam required by the main turbine to produce 28,000 kw. per hour when no steam is extracted from the turbine

X = the number of thousands of pounds of steam bled from the main unit to heat the feedwater in the first-stage heater

Y = the number of thousands of pounds of steam bled from the main unit to heat the feedwater in the second-stage heater

Z = the number of thousands of pounds of steam bled from

the main unit to heat the feedwater in the third-stage heater

r_x = the number of pounds of steam which have to be added per thousand pounds extracted from the x stage

T_c = the temperature in deg. fahr. of the condensate leaving the main condenser

T_x = the temperature of the condensate leaving a heater supplied with steam from the x stage

B = lb. of steam entering condenser

C = B.t.u. by test in the amount of steam required for sealing the high-pressure gland of the main unit. This is a constant, and is 5,500,000 B.t.u. above the liquid temperature of 78 deg. Where it is desired to use the number of B.t.u. in the gland steam above 32 deg. a figure of 5,700,000 is used

D = total number of pounds of live steam to be supplied to turbine

E = net kw-hr. put out by the plant

H_x = average total heat in the steam bled from the main unit at the x stage

F = equivalent number of pounds of steam that each boiler would evaporate if supplied with feedwater entering the 20-high boiler or the economizer in case boiler is equipped with economizer at 210 deg. fahr.

H_s = total heat in the steam leaving the superheater

h_x = total heat in the condensate above 32 deg. after being heated by the steam bled from the main turbine at the x stage

R = rise in the economizer after condensate has been heated by steam extracted from the one or more stages of the main unit

R_{210} = rise in the economizer when condensate enters economizer at a temperature of 210 deg.

B_e = boiler efficiency

P_e = ratio between results which we expect to get in ordinary operation as compared with the results which might be expected from the manufacturer's guarantees. This ratio includes the plant losses from condensation, steam leaks, soot-blower loss, radiation from steam pipes, etc.

G = total steam output of boilers in operation supplying

each unit. In this study each boiler was assumed to have an efficiency of 75 per cent when the boiler and economizer together were supplying 100,000 lb. of steam per hour or 78 per cent when steam was supplied by a 20-high boiler operating at the same output.

Total live steam to turbine $= A + Xr_x + Yr_y + Zr_z$

Amount of steam to be bled by first-stage heater

$$= X = \frac{(A - X - Y - Z + Xr_x + Yr_y + Zr_z) \quad (T_x - T_c) - C}{H_x - h_x}$$

Amount of steam to be bled by second-stage heater

$$Y = \frac{(A - Y - Z + Xr_x + Yr_y + Zr_z) \quad (T_y - T_x)}{H_y - h_y}$$

Amount of steam to be bled by third-stage heater

$$Z = \frac{(A - Z + Xr_x + Yr_y + Zr_z) \quad (T_z - T_y}{H_z - h_z}$$

To find the relation between X and Y by changing the form of the equations, we get

$$\frac{X(H_x - h_x) + C}{T_x - T_c} = A - X - Y - Z - + Xr_x + Yr_y + Zr_z$$

$$\frac{Y\,(H_y - h_y)}{T_y - T_x} = A - Y - Z + Xr_x + Yr_y + Zr_z$$

Subtract

$$Y = \left[X + \frac{X\,(H_x - h_x) + C}{T_x - T_c}\right] \frac{T_y - T_x}{H_y - h_y}$$

All the quantities in the above equation are known except X and Y so that a definite relation can be established between them.

Similarly by combining the expressions for Y and Z the relation between them will be found to be

$$Z = Y\left[1 + \frac{(H_y - h_y)}{(T_y - T_x)}\right] \frac{(T_z - T_y)}{(H_z - h_z)}$$

Substituting the known values for a combination of the fourteenth, eleventh and eighth stages we find that these equations when solved give simple answers.

$$Z = 0.771Y$$
$$Y = 1.110X + 5915$$
$$Z = 0.771Y = 0.855X + 4550$$

Substituting these values of Y and Z in the equation for X permits us to find X and as the relation of X to Y and Z are known we can then readily find the amount of steam bled at each stage and multiplying it by the ratio of the amount of live steam that has to be added per 1000 lb. bled we can find readily the total steam supplied to the turbine, as follows:

$$D = A + Xr_x + Yr_y + Zr_z$$

The steam passing to the condenser is

$$B = (D - X - Y - Z)$$

There are two variables in considering the boiler rating; one is the number of pounds of steam to be evaporated, and the other is the number of B.t.u. to be added, which varies on account of varying feedwater temperature. When the economizer was supplied with 210-deg. feedwater, each of the three 14-high boilers and economizer was assumed to be producing 100,000 lb. of steam or a total of 300,000 lb., the efficiency of the boilers alone were taken as 75 per cent (see Fig. 24) and the boiler efficiency under all other conditions was determined from this by correcting for changes in efficiency due to changed output and taking into account the effect of variation in feed temperature.

If F = the output of each boiler in pounds of steam per hour referred to 210-deg. feed temperature as a basis,

$$F = 100000 \times \frac{(H_s - h_z - R_z)}{H_s - h_{210} - R_{210}} \times \frac{D}{300{,}000}$$

Fig. 24 gives the efficiency of the 14-tube high boiler.

The net B.t.u. per kw-hr. output of the plant is obtained from the formula:

$$D \times \frac{(H_s - h_z - R_z)}{E} \times \frac{1}{B_c} \times \frac{1}{P_c}$$

where E is equal to the net kw-hr. put out by the plant.

Where the plant was equipped with economizers and induced-draft fan, and a motor-driven boiler feed pump was used, the plant auxiliary power requirements were assumed to be 1600 kw. and E was equal to 28,000 — 1600 or 26,400 kw-hr. Where a tur-

bine-driven boiler feed pump was used, the plant auxiliary power requirements were assumed to be 1400 kw. and E was equal to 28,000 — 1400 or 26,600 kw-hr.

In case the high boiler is used without the economizer, the rise in the economizer is neglected in the last two equations, and the boiler efficiency is taken from Fig. 25 instead of Fig. 24.

Where the high boiler is used with natural draft, the boiler feed pump requirements are reduced from 200 to 175 kw., and

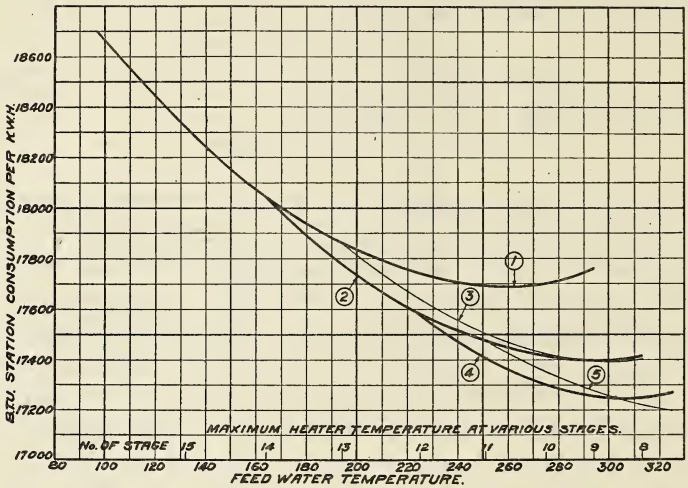

FIG. 28 EFFECT OF SINGLE-, DOUBLE- AND TRIPLE-STAGE EXTRACTION HEATING ON FEEDWATER TEMPERATURE AND STATION ECONOMY

North extension of Windsor power station. 30,000-kw, G. E. turbines. 28,000-kw. load. Steam at throttle 300 lb., 200 deg., 1 in. 20 high, cross drum B. &. W. boilers. No economizer, natural draft.

the auxiliary power is reduced 150 kw., on account of the omission of the induced-draft fan.

In case the steam-turbine-driven pumps are operated, Item C is increased by the amount of heat in the exhaust steam. In arriving at the B.t.u. in the exhaust steam allowance was made for variations in back-pressure by increasing the water rates of this unit ½ per cent per pound increase of back pressure.

A great many of the above factors are constant, and only a few change so that when one is familiar with the method the points on the curves can be obtained at the rate of two or three per hour.

Fig. 28 is a study of single-, double- and triple-stage heating

for the Windsor turbines, in combination with a 20-tube high boiler having an efficiency of about 78 per cent at the point at which it is operated.

Curve No. 1 shows the results obtained with single-stage bleeding.

Curves Nos. 2 and 3 show the results obtained with double-stage heating. There is apparently very little difference whether

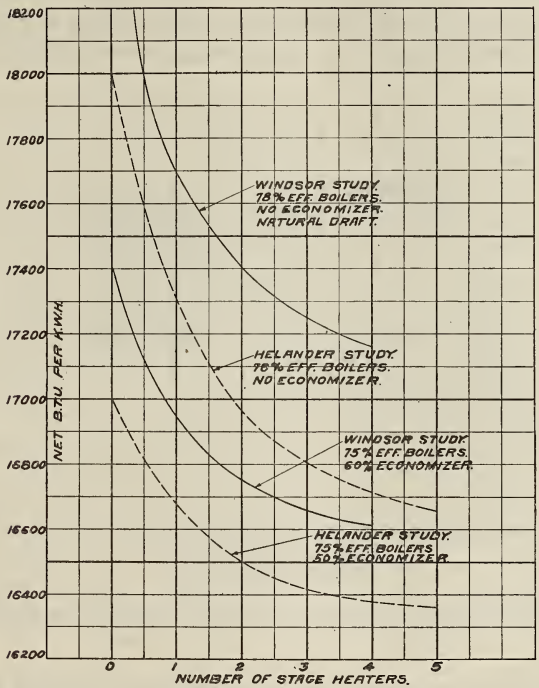

Fig. 29 Comparison of Windsor and Helander Study of Effect of Various Stage Bleeding on Power Station Economy

the thirteenth or fourteenth stage is used as the first stage.

Curve No. 5 shows the result obtained with triple-stage heating, the best results being obtained by a combination of the fourteenth, eleventh and eighth stages. It is possible that a higher stage might be slightly more efficient, but the data for the higher stages were not available.

Fig. 29 is a comparison of the results obtained by Mr. Helander for various stage bleeding, with the results of the Windsor study. Mr. Helander limited his study to four stages, but the

curve is projected so as to show the approximate results for five stages. The Windsor study was made for three stages, but is extended so as to indicate the approximate results for four stages.

Fig. 30 shows the temperature at which the best results were obtained for single-, double- and triple-stage bleeding in the Windsor study and single-, double- and quadruple-stage bleeding in the Helander study.

These studies clearly indicate the advantage of bleeding the main unit with or without economizers, there being a gain of 1.10 per cent of double-stage over single-stage heating and approxi-

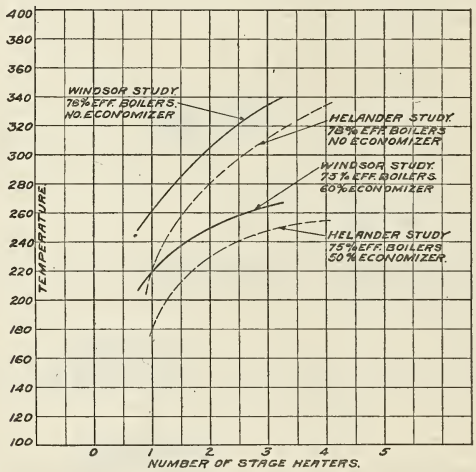

Fig. 30 Comparison of Windsor and Helander Studies Showing the Temperature at which best Results are Obtained for Various Stage Bleeding

mately one half that amount in addition if triple-stage is used in place of double-stage heating, that is, for a 14-tube high boiler equipped with economizer; whereas for a 20-tube high boiler there is a gain of 1.64 per cent in double-stage heating over single-stage, and an additional 1.00 per cent if triple-stage heating is used in place of double-stage.

Regarding reliability, while the heater condensers will complicate the condensate piping and increase the pumping head, with the possible exception of the effect of breakage of extraction heater tubes, it is difficult to see how they will affect the reliability of the plant or complicate the operating problems. The breakage of a condenser tube can be taken care of either by installing check valves

between the heater and the main unit or in the lower-stage heaters by putting in drip lines of large enough capacity to take care of possible leakage. Condensate that leaks through a broken tube in this way is returned to the condenser or condensate system, and is not lost. Gate valves should be installed between the heater and the main unit so that the heater can be disconnected from the unit if desired.

Regarding the capacity of condensing equipment, as the bleeding of the main unit reduces the amount of heat passing to the main condenser, some reduction in its size is permissible. Fig. 31

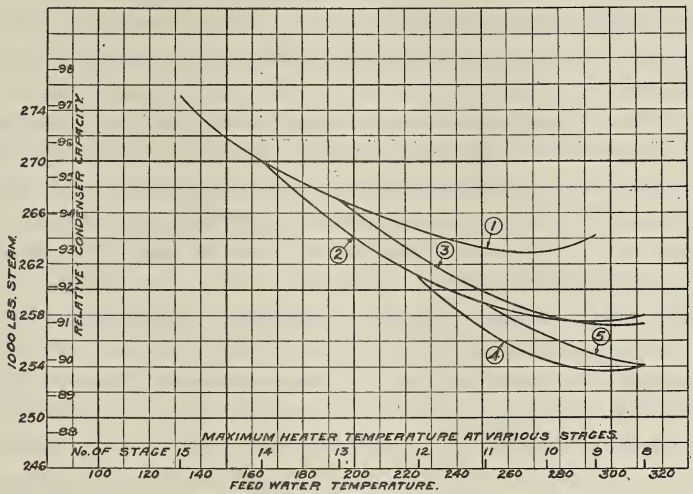

Fig. 31 Amount of Steam Passing to Main Condenser and Relative Size of Main Condenser Required

shows the pounds of vapor and condensed steam entering the main condenser per hour for various bleeding combinations. This study indicates that for single-stage bleeding the condenser need only be 93 per cent of that required if no steam is bled from the main unit. For double-stage heating this ratio becomes 91 per cent and for triple-stage heating 90 per cent.

Fig. 26 indicates that heating the feedwater by exhaust steam from the house turbine is much less efficient than by bleeding the main unit, there being an advantage of approximately 1.85 per cent in favor of bleeding the main unit. This has led to a change in the type of house turbine installed. The tendency seems to be to carry only as much load on the house turbine as is necessary

for the sake of reliability, paralleling the house turbine with the main unit and carrying all the load on the main unit, the switch being so arranged that in case of a heavy overload on the system, the house turbine with certain auxiliaries will pull away from the main unit and the house turbine will carry these auxiliaries at a slightly lower frequency until such time as the load can be again picked up by the main unit. The latest proposition is to carry no load on the house turbine but have it running so as to be able to pick up the necessary auxiliaries in case of trouble to the main unit. With this latter arrangement it is proposed to run a small pipe from the exhaust end of the house turbine to the condenser of the main unit, a check valve being placed in the main exhaust pipe from the house turbine, maintaining in this way a rarefied medium for the rotor to spin in, so that it will not overheat when running idle. The vacuum required in the exhaust end of the house turbine to prevent overheating varies with the design of the house turbine. It is only with the most efficient types of turbine that there is any danger that the vacuum which it is possible to maintain in the exhaust end of the house turbine will not be high enough during the warm summer months. The losses of such a stand-by house turbine when running in a high vacuum are very small.

It is possible to heat the condensate about 13 degrees by using the condensate to cool the air in a closed generator cooling system; and a rise of 7 degrees more may be obtained by absorbing the heat in the transformer and turbine oil. The use of condensate in these cooling coils will keep them clean; but there is some slight complication in regard to the operation of such a system during the warm summer months, or in case of dropping of load by the main unit. This latter is principally important in case the transformers and turbine are not paralleled as a unit on the high side of the transformers but paralleled with the other units on the low side of the transformers. These transformers will stand an interruption in the cooling water supply for several minutes without injurious effect. In studying such a system and comparing the reduction in heat requirements, consideration must be given to the fact that by absorbing this waste heat the amount of steam which can be bled from the main unit is reduced. While there is a possible reduction in the heat requirements of the plant of $1\frac{3}{4}$ per cent by absorbing the waste heat, if instead, additional steam is bled from the thirteenth stage of the main unit the heat requirements will be about 136 B.t.u. per net kw-hr. higher than if the

condensate temperature is raised by the waste heat in the generator air and transformer and turbine oil or the net gain in station economy of absorbing this waste heat is about eight-tenths of one per cent.

T. E. Keating. In view of the present agitation for improved efficiency in central stations, Mr. Helander's paper is quite timely. It is somewhat surprising to find that in the application of economizers, with 350-lb. boiler pressure and stage heating that the most efficient feed temperature is close to 200 deg., as it has been a commonly accepted belief that the efficient use of economizers demanded a somewhat lower temperature. If the desirability of this higher value can be proven, it will eliminate to a great degree various problems such as tube sweating, and if open heaters or surge tanks are used, the deaëration of water, which becomes increasingly difficult and expensive with lowered feed temperatures.

The use of multi-stage heaters is receiving considerable study in central-station engineering and the application involves many practical problems, such as the size and design of heaters, the heat transfer in event of superheated steam being bled at high pressures, the size of piping with low pressures, the use of traps and pumps, and provision for avoiding flooding the turbine. The problems of station layout seem to limit the practical application to four or five stages as Mr. Orrok states, and Mr. Helander's curve, Fig. 22, further indicates that with 350-lb. pressure the thermal gain is relatively small beyond three stages.

When a house turbine is used in combination with main unit bleeding, there is sometimes a question as to whether this auxiliary unit should be operated with approximately atmospheric exhaust pressure or at some pressure below atmospheric. Inasmuch as the efficiency of a commercial type house turbine at atmospheric exhaust is relatively poor, the greater thermal economy should occur with a house turbine expanding to a fair vacuum. However, if the last stage of heating is done in the main unit, the feed temperature will vary with the station load, while if done with a house turbine, the auxiliary load on the house unit may be adjusted so as to maintain a constant feed temperature. This same result could be obtained by use of a house turbine expanding to low vacuum, and in addition fitted with a bleeder valve for extracting steam at constant pressure. There is probably very little change in overall station efficiency due to the variable feed temper-

ature with changing load, and inasmuch as there seems to be a tendency to supply more rugged electrical equipment and more protective apparatus to insure continuity of energy supply to auxiliary feeders, the use of a house turbine as secondary source of power supply will probably become less frequent. Therefore, the house turbine should not be a controlling factor in the determination of the thermal efficiency of a station.

Mr. Helander's paper does not touch on the use of condensate for recovering the heat from the air used for cooling the main generator, presumably because he felt the thermal gain was not sufficient to warrant its use in a purely heat study. While the use of a closed cooler in connection with air recirculation is of great value, both in reducing the fire hazard and maintaining cleanliness of the generator, it is doubtful if the cost of the cooler and piping system to utilize condensate for this purpose can be commercially justified on a thermal basis when compared with obtaining the same amount of feed heating by bleeding the main unit.

F. H. Rosencrants. Mr. Helander's paper comes at a very opportune time in view of the fact that many new power plants under consideration propose incorporating feedwater heating systems designed along the lines discussed. The variety of schemes with reference to the number of stages of heating and the final temperature of feedwater proposed are indicative of the lack of harmony of thought among engineers on this subject. Such deductions as follow in this discussion are predicated on the figures as they stand in the paper with no attempt at revision or checking.

Mr. Helander points out that with an infinite number of stages of bleeding the cycle of power generation may be transformed from the Rankine cycle to the Carnot cycle, and the question arises, "What would be the corresponding ultimate saving?" Taking the author's assumed conditions of 330 lb. gage pressure and 200 deg. fahr. superheat at the throttle and a back pressure at the turbine exhaust of 1 in. Hg. absolute, the Rankine cycle efficiency figures out 35.35 per cent; the Carnot cycle (as modified by superheat) for the same conditions gives an efficiency of 40 per cent. The difference of 4.65 points in efficiency represents a saving of 13.15 per cent.

While the above efficiencies are those of the ideal cycles, it is probably fair to assume that each of the cycles may be approached in practice with equal perfection, and that, therefore, the actual

ultimate saving in practice will not be far from that calculated for the perfect cycles.

If we refer to Fig. 22 showing the B.t.u. per kw-hr. with various stages of feedwater heating, we will observe that one stage of heating gives a saving of approximately 3.8 per cent, 2 stages 5.8 per cent, 3 stages 6.8 per cent, 4 stages 7.3 per cent, and 5 stages 7.6 per cent. As Mr. Helander has already pointed out, the gain from multiplying stages rapidly diminishes with each successive stage added. From the above figures it would seem that three or perhaps four stages is about the limit for practical application, and that in view of the complications of equipment, piping, etc., it is doubtful if heating stages beyond three can be justified.

In all cases, the results in the paper are tied up with the use of house turbines. It would be interesting to draw comparisons of this type of installation with an installation in which all feedwater heating was accomplished with steam bled from the main unit. There is no question that the latter type of installation would be the more efficient, the difference between the two being a portion of the price we pay for the added reliability of auxiliary drive brought about by the use of the house turbine. We say " a portion of the price we pay," since in addition to the loss of efficiency the house turbine set also complicates the piping lay-out and station operation.

A point of particular interest brought out in Mr. Helander's paper is the effect of the efficiency of stage bleeding for feedwater heating on the application of economizers. In the past, there has been a tendency on the part of many engineers and manufacturers of equipment to ignore the less efficient transformation of power in the turbine room incident to the return of condensate to the economizer inlet cold. This in effect is to ignore the superiority in efficiency of the Carnot cycle over that of the Rankine cycle. These same engineers insist the gain of boiler and economizer efficiency as a result of this cold water as compared with hot water to be a net gain, and refuse to admit of an offset against this gain due to a reduction of efficiency in the steam cycle. With this idea in view, many engineers have insisted that for maximum economy the water should be returned to the economizers at the coldest possible temperature and that the practical limit was that temperature at which sweating of the economizers tubes, with consequent corrosion, begins to give trouble. Even with inefficient auxiliary drive turbines this was not true and with the much more

efficient use of heating steam as accomplished by stage bleeding of the main unit, it is more emphatically untrue. Summarizing from that portion of Mr. Helander's paper dealing with "Case 2," the following most economical temperatures with and without economizers are shown:

Heating stages	Most economical feed temperatures		
	Without economizers	With economizers	Difference
1	190	165	25
2	270	220	50
4	330	240	90

It will be observed that for multiple-stage bleeding the entering water temperature at the economizer is so high as to give but a small temperature difference between water and flue gas. An idea may be gained of the effectiveness of the economizers (50 per cent of boiler surface) by further summarizing from "Case 2" of the paper the B.t.u. per kw-hr. with and without economizers (both at the most economical feed temperature) and calculating the saving. The figures follow:

Heating stages	Heating B.t.u. per kw-hr.		
	Without economizers	With economizers	Difference
1	17370	16700	670
2	17050	16680	370
4	16850	16600	250

$$\frac{17370 - 16700}{17370} = 3.8 - 0.50 = 3.30 \text{ per cent}$$

$$\frac{17050 - 16680}{17050} = 2.2 - 0.50 = 1.70 \text{ per cent}$$

$$\frac{16850 - 16600}{16850} = 1.5 - 0.50 = 1.00 \text{ per cent}$$

0.50 has been deducted from the calculated gross gain as the allowance for the power requirements of the induced draft fan. In stating these figures attention should be called to the fact that a

boiler efficiency of 78 per cent without economizers and 75 per cent with economizers has been assumed. Accepting the above figures as representative of what may be actually expected, it gives us cause to question the practicability of combining economizers with multiple-stage bleeding and in view of the fact that the multiple-stage bleeder heater system, plus additional boiler capacity is likely to be less expensive than the economizer installation, it seems proper to ask the question, "Has an economizer a place in the central station of modern design?"

If we may for the moment assume the economizer eliminated, the field is left open for the application of air pre-heaters. There is, however, at the present time little data available on the cost of installation or on the operation of such equipment, though there are a number of installations for which air pre-heaters are being considered and if adopted will furnish interesting material for future papers.

An added advantage of the closed-heater system incident to the application of heating in stages with bleeder steam not touched upon in the paper is that the condensate at all points after leaving the condenser is under pressure and therefore the opportunity for the condensate to absorb air, which will later result in the corrosion of piping, economizers, boilers, etc., is eliminated. If we may assume a temperature of condensate in the economizers to be within two or three degrees of vacuum temperature, a figure which is being reached in condensers of modern design, the condensate upon leaving the condenser should show very small air content. It only remains to deaërate the make-up supply before admitting it to the system to obtain feedwater to the boiler plant in practically an air-free condition.

Oscar F. Junggren. The design of power stations is, of course, a compromise like many other things, but it is well to take stock every once in a while to see what is the ideal condition we wish to obtain. In reality, the ideal condition is to produce power most economically and this will always be done in the main unit. Whether the power is consumed by auxiliaries or sold, the most economical arrangement is to produce the power in the main unit and to heat the feedwater by bleeding. It has been felt that not all auxiliaries can be motor driven, but steam-driven auxiliaries should be reduced to a minimum. By bleeding the turbine we have the advantage of reducing the congestion in the low-pressure end

of the turbine. The modern tendency is toward motor-driven auxiliaries, permitting the highest efficiency in the production of power.

Francis Hodgkinson. Stage heating of a main unit for the purpose of heating boiler feed is a reversion to ancient practice. It seems curious that we have been so slow to adopt it in modern steam turbine power house practice.

I think all turbine manufacturers realize the saving which will be the result of this practice, and later design large turbines built by the Westinghouse Electric & Mfg. Co. are provided with means for such heating. Four connections are generally provided on the turbine, where, at the point of best steam consumption, the temperature will be 140 and 200 deg. fahr., and 50 and 120 lb. per sq. in. absolute, these points being subject to slight adjustment, if necessary.

I am surprised that the author laid so much stress in his paper on the application of the so-called house turbine, for less economy is to be secured by its use, as compared with stage bleeding of the main unit. The house turbine was employed as a substitute for separate steam driven auxiliaries and with the idea that, inasmuch as this house service system would be independent of the rest of the plant, reliability of the auxiliary service would be secured. However, it is entirely possible to secure at least as high a degree of reliability in other ways, as, for instance, operating the auxiliaries from the main bus bars through a motor generator set, which motor generator set has a simple turbine connected to it which would run idly, having its governor adjusted so that it will come into operation whenever there is a fall in frequency. It has been suggested that this turbine exhausts to the atmosphere when operating under steam through a non-return free exhaust valve, there being a small connection to the main condenser, so that when the turbine is running idly, it will be under the same vacuum as the main turbine for the purpose of reducing the windage loss.

Another alternative would be an additional generator driven by the main turbine solely for the operation of the auxiliaries.

I shall be surprised if house turbines are employed very much in the stations of the future.

Nevin E. Funk. Theoretically, stage bleeding is probably the most efficient method of heating feedwater, but the writer is not quite sure but that it lends itself to more complication than is found

in other means of obtaining exhaust heat. Admittedly the main unit is most efficient and the heat obtained from it for feedwater heating is not thrown away in the circulating water and is obtained with a better water rate than is possible with auxiliaries. However, when things go wrong in a power plant, it is well not to have everything so connected that all operation must cease. Simplicity makes for efficiency. Many schemes which look feasible on paper will not work in practice because of the human element which is always involved.

C. M. Hardin.[1] Everyone concerned with steam power generation realizes that reliability is of first importance and must be obtained even at the sacrifice of plant efficiency and other requisite factors.

Without discounting reliability, it is, however, vitally important that high efficiency be obtained. To obtain maximum plant efficiency every integral part of the station equipment must be of such correlative design and construction as will individually produce and maintain high efficiency.

Stage bleeding from the main prime mover is without question the method that does give highest feedwater heating efficiency as compared with using the house turbine or other steam driven auxiliaries. If stage bleeding should even in the slightest degree tend toward unreliability as regards continuity of service, the problem is to perfect the equipment used and allied with the scheme.

Advancement in any art can only be procured by development and by strengthening all links in the chain. Therefore, rather than advancement, which is vitally necessary, do not stick to ease and conservatism by adhering to old and less economical practices.

The Author. The value of recovering low temperature heat, such as can be obtained from the ventilating air used on generators, is dependent principally on the temperature to which the feedwater is heated in the first-stage heater and the efficiency of the method of heating the feedwater. Except where other factors than that of fuel economy influence the layout, investments, such as at present appear necessary to recover the equivalent of the generator losses, would seem to be of speculative value. The absorption of heat by the feedwater before it enters the first-stage heater requires that the steam bled from the main unit be reduced

[1] Ross Heater & Mfg. Co., Inc., 2 Rector St., New York, N. Y.

by an amount equivalent in heat value to the heat absorbed, and this largely offsets the gain from the recovered heat. The curve of Fig. 32 shows in per cent of the recovered heat the reduction in the heat consumed by the generating station obtained by recovering low-temperature heat. With single-stage heating and a feedwater temperature of 150 deg., the heat regained is seen to be about $\frac{1}{3}$

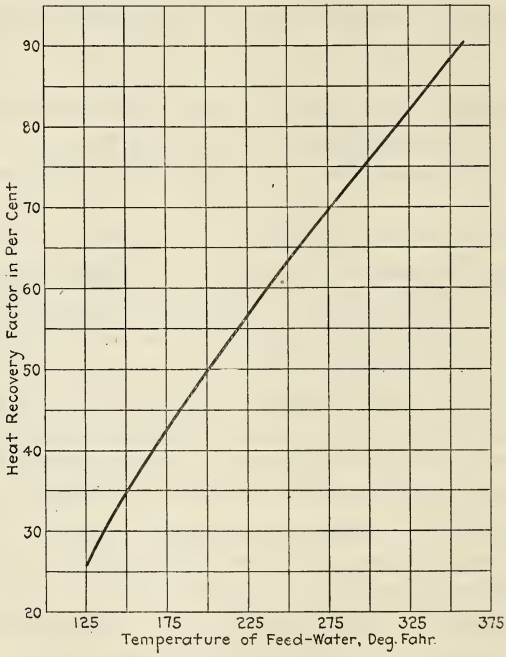

FIG. 32 FACTOR TO DETERMINE VALUE OF RECOVERING LOW-TEMPERATURE HEAT WITH SINGLE-STAGE HEATING

When low-temperature heat is recovered, as is done when the ventilating air of the main generator is cooled by condensate, the reduction in the heat consumed an hour by the generating room may be approximated by multiplying the quantity of heat recovered by the heat recovery factor shown by the above curve. For accurate analyses, exact data based on the bleeding characteristics of the turbine and the method of heating the feedwater should be used to determine this factor.

of the heat absorbed by the feedwater. If the generator losses are taken as 800 kw. for a 30,000-kw. generator, operating at 25,000-kw. load, the heat regained per hour is 270 kw-hr. and this, with a generating room thermal efficiency of 25 per cent, is equivalent to 68 kw. at the switchboard. If the fuel cost of power is $\frac{1}{2}$ cents per kw-hr. the gain in money will be 34 cents per hr. or $2,980 a year

of 8,760 hr. This latter figure assumes a use factor of 100 per cent and a load factor of 100 per cent. Inasmuch as the combined use and load factor probably will not exceed 50 per cent, the net return in money value by recovering the generator losses will probably not exceed $1500 a year, which with 15 per cent fixed charges warrants an investment of not over $10,000. With multiple-stage heating the saving will be less.

The comments regarding the elimination of the house turbine indicate a healthy desire for both simplicity and economy of operation. Justification for the use of house turbines, in so far as these have been justified, lies in the substantial insurance they provide against disruption of the auxiliary power supply, and this insurance has been knowingly purchased with a sacrifice in economy. Even with large and interconnected stations, if the auxiliary power is derived directly from the main bus, faults on the system which cause the frequency to drop may entail a complete shut-down of the station due to the consequent slowing down of the condenser circulating pump. Some source of independent power supply is therefore required for the essential auxiliaries and particularly for the circulating pumps on the condenser of the main unit.

A house turbine of sufficient capacity to supply power to the essential auxiliaries is an altogether adequate safeguard against complete disruption of service, and as such the reduction in the station economy due to its no-load losses are not of great concern. For a 650-kw. house turbine, the no-load losses would be approximately equivalent to 65 kw. which would seem a small expenditure for insurance in a station developing 25,000 kw. The suggested method of floating a house turbine on the line without load but available for emergency service would probably mean the use of units with at least an equal, if not greater, no-load loss. The stand-by house turbine probably would be of sufficient size to carry the entire auxiliary load or for the case covered by the paper, 1300 kw. If, due to special design, the no-load losses are made half of those of an ordinary house turbine of equal capacity, the no-load losses would be 65 kw. or the same as that of a 650-kw. house turbine. If, however, a motor generator set is used in conjunction with the stand-by turbine, the combined losses of the turbine and motor generator set would be approximately 165 kw. or 100 kw. more than those of the 650-kw. house turbine. The comparison between these methods should be based, however, not upon the no-load losses but rather upon the relative steam con-

sumptions, in the one case running a house turbine under load and in the other case running it without load.

On the basis of such a comparison the method of driving the house turbine without load would no doubt show an improvement in economy over that obtainable when the house turbine is loaded, provided a motor generator set is not used. If a motor generator set is used in connection with the stand-by turbine it is quite likely that the steam consumption will be approximately the same as that required when using a 650-kw. house turbine. The efficiency obtainable in favor of the stand-by turbine not loaded is due to more efficient utilization of steam in the main unit than in a house turbine. From the point of view of capital charges, however, the house turbine should be credited with being a power unit, equal in money value to the cost of its equivalent in main unit capacity.

Stations with four or more main units might advantageously use house generators coupled directly to the main unit shafts, interconnecting these generators. These auxiliary house generators would be preferably of sufficient capacity to carry the essential auxiliaries such as the circulating pump, and, in addition, to provide power for driving the circulating pump of a unit about to be started or having an auxiliary generator out of service. The other auxiliaries could then be carried by main unit power. Some reactance to compensate for phase shifts between the auxiliary generators would probably be required, and, as the generators will have slower speeds than house turbine generators, investment charges would have to be included in a proper comparison between the two methods of supplying auxiliary power. This arrangement from the point of view of fuel economy would no doubt have advantages over the house-turbine layout. When the station has but two or three units in operation, the auxiliary power bus will no doubt be interconnected with the main bus for starting-up purposes and to provide an additional source of power to replace that of a non-operative auxiliary generator. Otherwise heavy reactances between the auxiliary generator and oversized generators would be necessary. Interconnection with the main bus, however, is a recognized hazard.

Probably the simplest and, in some measures at least, the most economical method of eliminating the house turbine is to provide dual driven circulating pumps with the turbine running idle but available for immediate service should the auxiliary power supply be disrupted. A 300-kw. turbine would approximately meet the

power requirements of the average circulating pump of the condenser of a 25,000- or 30,000-kw. unit. This 300-kw. turbine would have small no-load losses, probably not exceeding 15-kw., and would give adequate assurance that the circulating pump would not be out of service. Objection to the arrangement would lie in the unsatisfactory piping layout that would probably result.

In the end it appears likely that engineering judgment will determine the method of providing reliable auxiliary power, and though it appears that the house turbine, as it now stands, will be modified it is likely that, for some engineers at least, this modification will take the form of reduced size rather than complete elimination. The main consideration is not economy but reliability of service. In any case, the effect on the heat balance and on the feedwater temperature of choosing between the various methods available for providing insurance against disrupted auxiliary power will not be sufficient to warrant changing the feedwater temperature as already determined. Mr. Bell has very completely worked out heat balances involving all the various methods of driving the auxiliaries and from his results an estimate of the advantages of the various methods may be obtained. It should be observed, however, that Mr. Bell's analysis is based on a house turbine capable of carrying the entire auxiliary load.

As Mr. Rosencrants points out, the improvements in economy obtainable by using economizers is reduced by stage heating. Whether economizers can be justified or not depends largely on the relative cost of boiler surface and economizer surface. Recent quotations indicate that economizer surface costs less per square foot than boiler surface and if this relationship holds when building and other construction charges are included, it is altogether possible that small gains in commercial economy may be had by using economizers. The value of one per cent improvement in economy using economizers is small but would be worth obtaining if the investment charges are not increased beyond that justifiable. This investment would be determined by the price of coal and so would vary with the locality.

Mr. Bell's discussion has the added value of being based on data obtained from manufacturers' guarantees and from the layout of an operating station. Mr. Bell, after determining the heat consumption rate of his station from manufacturers' guarantees, divides this by 95 per cent to determine his station B.t.u. rate and this largely accounts for the differences in the heat required to

produce a kw-hr. as determined by the author and Mr. Bell. The author took into account leakages and make-up losses but these were not equivalent to the use of a factor such as 95 per cent for correcting guarantees to operating conditions. The temperature of the feedwater entering the first-stage heater used by Mr. Bell was 97 per cent while that used by the writer was 75 per cent. Radiation and leakage losses in the author's study were considered

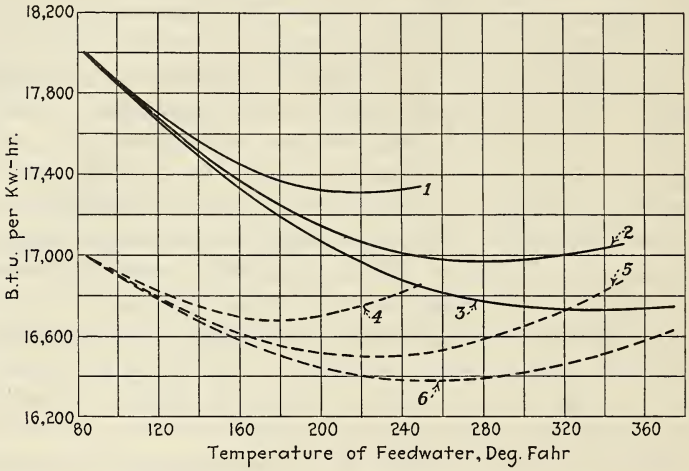

Fig. 33 Heat Consumption per kw-hr. Neglecting Leakages from Low Pressure Steam Piping

The full line curves 1, 2 and 3 are for single-, double-, and quadruple-stage heating respectively, without economizers. Curves 4, 5 and 6 are the same except that economizers of 21,850 sq. ft. area, equivalent to approximately 50 per cent of the boiler area, were used.

The temperature of the feedwater for best efficiency is seen to increase with the number of stages.

The curves are rather flat in the vicinity of best efficiency so that ordinary deviations from the best feedwater temperature will not affect materially the economy.

For comparative purposes the same economizer area was used for curves 5 and 6 as for curve 4, though commercial considerations would warrant reducing the economizer area for double- and quadruple-stage heating to 18,150 sq. ft.

as increasing with the temperature of the feedwater and this would tend to establish a lower feedwater temperature than the method used by Mr. Bell. Taking these matters into account the agreement between the feedwater temperatures determined by Mr. Bell and the author is close. The discrepancy between the temperature obtained by Mr. Bell and the author appears to be about 20 per cent more at the single-stage point than at the other points. By reference to the curves of Fig. 33 which show the summary of the

author's data on the same basis as that prepared by Mr. Bell, except for the initial temperature of the feedwater and the method of accounting for radiation and leakage losses, it is evident that this difference is not of considerable concern due to the flatness of the heat consumption curves. For this set of curves, leakage losses from the low-pressure steam piping were ignored and the same size of economizer was used throughout.

No. 1874

THE COMMERCIAL ECONOMY OF HIGH PRESSURE AND HIGH SUPERHEAT IN THE CENTRAL STATION

By Geo. A. Orrok, New York, N. Y.
Member of the Society

This paper is an attempt to examine into the theoretical relations between efficiency and pressure and temperature. During the investigation all of the available experiments on the variation of the strength of materials with temperature have been examined and certain inferences have been drawn. The variations in cost have been approximately evaluated and the conclusions of the discussion are stated in the summary.

IN THIS AGE of progress when the designers of central stations of all types and kinds are vying with each other in the use of higher pressures and temperatures as well as multiplying the complication of the steam generator, prime mover, and auxiliaries, it is expedient to go back to the beginnings of the central station and by a review of the line of development get a broad perspective of the field of power generation which will enable us to apply, to the newer developments, those basic principles of commercial economy and efficiency necessary to a proper solution of the problem.

2 The earlier use of steam when its expansive force had not been discovered, is best shown by the Newcomen pumping installations, in which the steam was used at atmospheric pressure. Watt discovered the expansive force of steam and so improved the boiler, engine, and steam piping that 15 lb. gage could be used economically. From his time until about 1900 rising pressure kept pace with improvements in materials and in the design and construction of steam generators and piping, with occasional excur-

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sions in the realms of higher pressure and temperature. Practice at this date (1900) may best be illustrated by the work of the Wildwood engine. Since 1900 improvements in steam generators, piping, and prime movers have been made with increasing rapidity and many modern plants are running on 250 lb. and 200 deg. of superheat using standard piping and valves with the newer designs of boilers, superheaters and turbines. Table 1 shows the pressures and economies theoretically possible and attained in a central station, with all the results reduced to British thermal units in the coal per kilowatt-hour of useful work.

3 Our engineers are now using better and more uniform materials than ever before. Our metallurgical and manufacturing methods have been greatly improved. Copper, wrought iron, and cast iron have been discarded for boiler purposes. Steel and even

TABLE 1 STEAM PRESSURES AND ECONOMIES THEORETICALLY POSSIBLE AND ACTUALLY ATTAINED IN CENTRAL STATIONS

Year	Engine	Pressure, lb. per sq. in. abs.	Temperature, deg. fahr.	Thermal efficiency, Carnot cycle, per cent	Actual B.t.u. per kw-hr.	Actual overall thermal efficiency, per cent
1770	Newcomen.....	15	212	2.7	416,000	0.82
1810	Watt.....	30	300	9.17	61,000	5.60
1900	Wildwood.....	215	400	20.8	19,200	17.75
1922	Modern.....	265	650	35.00	18,000	19.00

alloy steel are in common use and their uniformity leaves little to be desired. Certain of the non-ferrous alloys are in common use and play their part in aiding the common security which characterizes modern installations. Copper boilers and 212 deg. maximum temperature gave place to wrought-iron boilers and 30 to 40 lb. pressure with a maximum temperature of 300 deg. The use of steel as a boiler material raised the pressure to 200 lb. and the temperature to 400 deg., while our modern steels allow pressures up to 400 lb. with a maximum temperature of 750 deg. In certain distillation plants temperatures as high as 1100 deg. have been obtained, but the life of the material is short.

4 Now it is well known that steel when heated to comparatively low temperatures, say, 900 or 1000 deg., loses its strength and becomes unfit to sustain loads and the heat strains from even moderate heating of 300 or 400 deg. may cause certain deforma-

tions of a highly unsatisfactory character. Cast-steel valves have deformed at 750 deg. to such an extent as to render them useless, and the "growth" of ordinary cast iron at temperatures above 450 deg. is known to be destructive, but notwithstanding this the later constructions at 250–300 lb. pressure and 700 deg. maximum temperature are commercial. The first cost is not excessive, repairs are moderate, and the life of the installation is all that can be desired.

COMMERCIAL LIMITS OF PRESSURE AND SUPERHEAT IN THE CENTRAL STATION

5 What, then, are the commercial limits of pressure and superheat in the central station? In Fig. 1 we have calculated

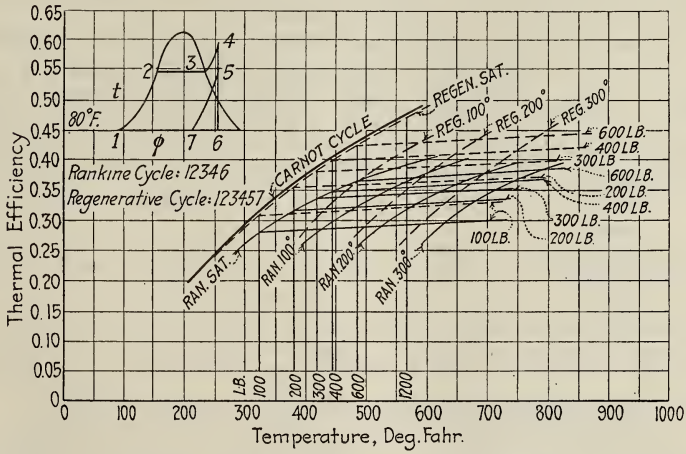


Fig. 1 Theoretical Thermal Efficiency of the Carnot Cycle, the Rankine Cycle, and a Regenerative Cycle, Using Both Saturated and Superheat Values for Ranges up to 1200 lb. and Superheats of 750 Deg. and Above

and plotted the theoretical thermal efficiency of the Carnot cycle, the Rankine cycle, and a regenerative cycle, using both saturated and superheat values for ranges up to 1200 lb. and superheats of 750 deg. and above. It will be seen that the chosen regenerative cycle for saturation follows very closely the efficiency of the Carnot cycle, while the Rankine cycle falls below the Carnot cycle increasingly with rise of pressures. The superheat lines for both

cycles are nearly parallel and maintain this characteristic over a wide range. The gain from regeneration increases with pressure and is constant for equal superheat. For any final temperature the gain from pressure increments decreases, but the gain from regeneration increases with pressure increments. Thus at 750 deg. final temperature the Rankine efficiency increases from 35.5 per cent to 40.5 per cent with pressure rise from 200 to 1200 lb., while the regenerative efficiency increases from 37 per cent to 48.5 per cent with the same pressure increase.

6 Fig. 1 dealing with theoretical efficiencies only, does not show the station losses, which aggregate around 45 per cent. Grouping the losses, we may say that the generator efficiency is 97 per cent; turbine efficiency, 78 per cent; boiler efficiency, 80

TABLE 2 SAVINGS IN B.T.U. PER KW-HR. EFFECTED THROUGH INCREASE IN STEAM PRESSURE

Case	Cycle	Pressure, lb.	Max. temp., deg. fahr.	Theoretical efficiency, per cent	Station efficiency, per cent	Actual efficiency, per cent	B.t.u. per kw-hr.
1	Rankine.....	200	750	33.5	55	18.4	18,500
2	Rankine.....	1200	750	40.5	55	22.3	15,300
	Saving.....	.....	.....	21	.....	.....	3,200
3	Regenerative....	200	750	37	55	20.35	16,800
4	Regenerative....	1200	750	48.5	55	26.7	12,800
	Saving.....	.....	.....	33.3	.....	.....	4,000

per cent, leaving for piping, auxiliaries, and radiation 91 per cent; all of which multiplied together give 55 per cent efficiency.

7 Table 2 shows how these figures work out for the above suppositions. The saving by regeneration (Case 1 — Case 3) is 1700 B.t.u., and in (Case 2 — Case 4) is 2500 B.t.u. Considering the trend of the curves in Fig. 1, comparatively little is gained by further superheating, but there has been no attempt to go above 800 deg. and the flattening out is much more marked as the temperature increases.

8 Water has its critical temperature at 704 deg. and the critical pressure is about 3200 lb. Pressures up to this limit may therefore be considered and since oil-still temperatures up to 1100 deg. have been used, we may consider superheat temperatures up to that point. We know that the first allotropic change in steel occurs at about 1300 deg., and 1100 deg. is well below this point.

Here, however, we must look into the physical properties of the steels used in power-plant construction.

9 When steel is heated the first sign of change — a dull red just visible in the dark — appears at about 750 deg. Visibility in daylight begins at about 850 to 900 deg. and a full dark red is attained at about 1100 deg. The full cherry red is attained at the first allotropic change of 1300 deg. Good boiler steels increase in tensile strength up to about 600 to 700 deg. and lose much in ductility, but above 800 deg. the tensile strength rapidly falls off and the ductility largely increases. Cast steels of proper carbon content and suitable for fittings, valves, and turbine construction show the same properties within rather narrow limits. Pipe steels have nearly the same characteristics, but the temperature at which the tensile strength starts to fall is around 550 to 600 deg. This lack of ductility or increase of brittleness at the maximum temperature of use must be compensated for by a larger factor of safety (i.e., increased thickness and weight), with higher cost and a more careful selection of material. But with increased thickness it must not be forgotten that the safe stresses must be correspondingly lowered. The variation of strength of materials with temperature is covered in Appendix No. 1.

VARIATION OF STRENGTH OF MATERIALS WITH TEMPERATURE

10 This subject may be considered from five points of view, depending on the conditions under which the materials are to be used.

11 *Boiler Material.* Here we know that maximum temperature cannot exceed 704 deg., the critical temperature of water, and that the fire side of the tube or drum can only be a few degrees hotter. If 1200 to 1800 lb. per sq. in. be the chosen pressure, the water temperature is 625 deg. at maximum and the fire side will not exceed 725 deg. normally. This temperature is well within the maximum-strength zone and only care is needed to secure sufficient ductility.

12 *Superheater Material.* The maximum temperature of the inner surface of the tubes may be a little higher than the maximum superheat. The outer temperature depends on the position in the boiler. Radiant-heat superheaters are now commercial where the pressure parts are protected with a heat-absorbing and conducting covering of cast iron. The fire surfaces may be 1400 deg. to

1600 deg. while the inner-wall temperature is as low as 700 deg., but 1100 deg. is comparatively easy of attainment. Bare-tube superheaters would have a maximum outside temperature of around 1200 deg., but the life at this temperature is unknown. The maximum safe point may be taken as 800 deg. superheat temperature and 950 deg. outside temperature.

13 Neither the tube manufacturers nor superheater manufacturers are afraid of these conditions, and I believe they are prepared to guarantee their product under such conditions.

14 *Piping Material.* Under this category the condition is markedly different. The steam side of the pipe is below the maximum temperature of the steam. The outer surface is much lower and the thickness of the material can be increased to compensate for the lower elastic limit of the material. The ductility is improving with temperature increase. Pipe joints are the only serious trouble, and the modified Van Stone joint with the pipe edges welded for tightness is the apparent solution. Experience with this joint has been satisfactory.

15 *Valves and Fittings.* Valves and fittings must necessarily be made of castings and the low-carbon, open-hearth steel used, while falling off in strength sooner than the worked material used in boilers and piping, has still a respectable elastic limit at 1100 deg. As in the last category, the inside wall temperature is lower than the steam temperature with still lower temperatures at the outer wall. But the shape of casting is all-important. Globe valves of the double-beat or Wanick type can be made with practically no flat surfaces and with two axes of symmetry. Internal pressures do not seriously deform this design, but external strains may cause minor troubles. Throttle valves and marine stop valves are usually of this type and can be kept tight at 750 deg. It is probable that a temperature of 1100 deg. could be safely undertaken with these valves if special precautions were used in the design of the seat, disk, and stem. Flexible-disk, double-beat valves are used in Europe at temperatures in excess of 750 deg.

16 Gate valves have only one axis of symmetry and flat surfaces of considerable size. Here seat and disk troubles are encountered at temperatures above 700 deg., and it may be some time before the manufacturers can guarantee a tight job at the higher temperatures. This type of valve is peculiarly sensitive to outside strains tending to deform the seats and disk.

17 *Turbine High-Pressure Ends.* Pressures can make little difference here as the high-pressure parts are comparatively small and without doubt the pressure will be reduced 50 per cent in the first nozzle. There need be no high-pressure stuffing box, so that the highest pressure to be packed against will be about half the maximum. Temperature strains will be the only thing that need be considered, and the expansion will reduce the superheat in the first-stage nozzle to a workable valve, which is 750–850 deg. It is to be noted that steam-chest or nozzle-box troubles have been the only prominent troubles where 400 lb. and 750 deg. superheat have been tried out, and it thus appears that a proper design will obviate most of the trouble. Cast or forged steel must be the material used.

18 In general, the turbine designers say that 1200–1500 lb. pressure has no terrors for them and can be used when desired. They can supply the apparatus. Temperatures are more troublesome, and no one apparently cares to go much above 700–750 deg. at the present time. Superheater and valve materials at present seem to be the deciding factor in the use of high pressures and temperatures, and there is apparently very little which careful design and the selection of proper materials will not overcome.

19 Only a portion of the central-station installation is affected by high pressures and temperatures, especially where electric auxiliaries are used. The boiler and piping system and prime mover cover the entire list, and in the usual central station this has represented about 40 per cent of the entire installation cost. Latterly, with the larger station the percentage is around 30, and may be taken at 25 per cent for the new and larger stations of which Hell Gate and Calumet are types. If the station cost be taken at $100 per kw. as an average figure this portion of the installation has cost $25, and at 15 per cent the fixed charges are $3.75 per year, or on 5000 hours' use 0.07 cent per kw-hr. What then, will be the extra cost of this apparatus when designed for higher pressures and temperatures? Standard boilers today can be bought up to 275 lb. pressure; 300 lb. necessitates thicker plates and tubes, and prices advance accordingly; 400-lb. boilers have been purchased recently and 500-lb. boilers have been built. Flash boilers of the Serpollet and De Laval types using pressures in excess of 1200 lb. and coil boilers of the Herreshoff type using equally high pressures have been built as experiments or in toy sizes as for automobile work or for high-speed launches. Schmidt's water-

tube boiler working at 900 lb. has been run 14,000 hours with marked success. Boilers for 400 lb. pressure have been offered at about 50 per cent increase in price and 1200-lb. boilers have been figured at a 100 per cent increase in price.

20 Turbine designers are at work on designs arranged to utilize higher pressures, but higher temperatures seem more troublesome to them. I have no estimate, but we can safely apply the same percentage increases of price to the turbine that obtain with the boiler. Steam piping decreases in size with increasing pressure, so much so that it is doubtful if a 1200-lb. steam line would cost more than a 200-lb. line of the same capacity. The volume of high-pressure steam at 1100 deg. maximum temperature is quite uncertain, but the chances are that our figures are not more than 10 per cent from the truth.

SUMMARY

21 We may then summarize as follows:

a Pressures up to 1200–1500 lb. at least are commercial and may be attained without serious difficulty

b Temperatures for the present are commercial up to 700–750 deg., which should not be exceeded until our materials for valves and superheaters are improved. More extended experience is necessary before the range of 800 deg. and over is attempted in a commercial installation

c With the completion of the Steam Table Research it will be possible to calculate accurately just where the possible limits of economy in pressure and temperatures may be located. At the present time the uncertainty of all values above 200 lb. pressure and 400 deg. fahr. temperature renders most of our calculations of only academic value

d There still appears to be more economy in a closer study of operation and construction losses — resulting in improving the efficiency of central stations installed in accordance with the present accepted canons of design — than in the attempt to increase widely the temperature range of our heat cycles. While we may save 4000 B.t.u. per kw. by using a regenerative cycle and increasing pressure and temperature, it should be pos-

sible to save nearly as much by a reduction to the lowest limits of the heat losses which we know exist in our present installations.

22 In Table 3 we have given for 700 and 1000 deg. maximum temperature and for four pressures the theoretical efficiency of the three cycles used, the practical efficiency of the Rankine and regenerative cycles, and the savings per kilowatt-hour for all con-

TABLE 3 SAVINGS IN COAL

									Saving over R. 200 lb.–700 deg., B.t.u. per kw.		Saving in lb. of 13,800-B.t.u. coal per kw-hr.	
Pressure, lb. per sq. in.	Superheat temp., deg.	Carnot effy., per cent	Reg. effy., per cent	Rank effy., per cent	Reg. × 0.55	Rank. ×0.55	Reg. B.t.u.	Rank. B.t.u.	Reg. C.	Rank. C.	Reg. C.	Rank. C.
200	700	53.5	37.0	33.4	20.3	18.35	16,820	18,620	1800	....	0.13	.....
400	700	53.5	41.2	36.3	22.6	19.95	15,100	17,120	3520	1500	0.255	0.109
600	700	53.5	43.5	38.0	23.9	20.9	14,300	16,350	4320	2270	0.313	0.1645
1200	700	53.5	47.5	40.0	26.1	22.0	13,100	15,500	5520	3120	0.40	0.226
200	1000	63.0	38.0	35.0	20.9	19.25	16,350	17,750	2270	870	0.165	0.065
400	1000	63.0	42.8	37.6	23.5	20.7	14,500	16,500	4120	2120	0.299	0.1535
600	1000	63.0	45.0	39.5	24.7	21.7	13,820	15,720	4800	2900	0.348	0.21
1200	1000	63.0	49.0	41.5	26.9	22.8	12,700	14,970	5920	3650	0.43	0.257

ditions over the Rankine cycle at 200 lb. and 700 deg. maximum temperature in B.t.u. and lb. of coal of 13,800 B.t.u. content.

23 These savings in cents per kilowatt-hour are plotted in Fig. 2 on the basis of $6.00 coal, together with the extra-fixed-charges curve. At the top of the figure are two curves showing the percentage increase in cost of boiler and turbine and the decrease in size of pipe required for a given quantity of steam. It will be seen that the extra fixed charges are always greater than the coal saving for the Rankine cycle (ordinary operation) at 700 deg. maximum temperature, and that the additional gain for the 1000 deg. maximum temperature barely exceeds the additional fixed charges. We cannot then expect much from increasing pressures and the ordinary methods of using steam. Increasing temperature helps to some extent but not enough to be attractive, since the saving is nearly all used up by the additional fixed charges.

24 With the regenerative cycle the results are much better and useful savings are indicated, increasing considerably with pressure and slightly with temperature. It is evident that some such regenerative cycle must be used, but the rewards from increasing temperature are not commensurate with the costs. Many regenerative cycles are possible, but the one used shows nearly the

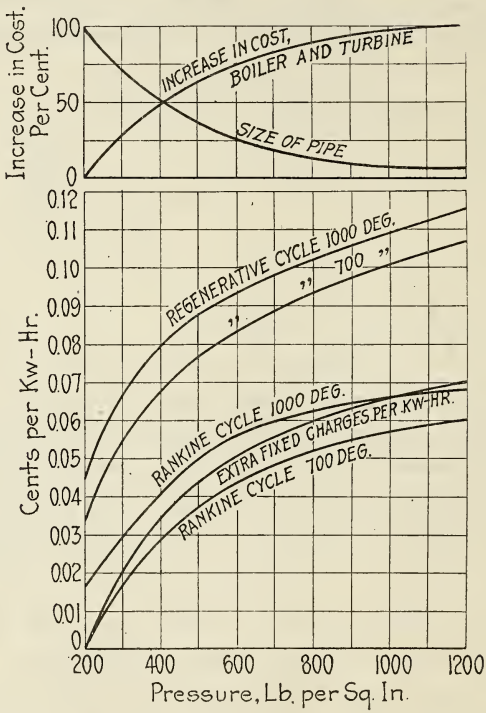


Fig. 2 Saving in Cents per Kilowatt-Hour for All Conditions Over the Rankine Cycle at 200 lb. and 700 Deg. Maximum Temperature, Based on $6.00 Coal

maximum economy. Such styles need many heat interchanges and a number of separate feed pumps for the close approach to theoretical figures. Indeed, Thurston and Stanwood[1] in 1899 pointed out that with an infinity of heaters and pumps the Carnot efficiency might be equaled. So far, however, four or five stages have been the ultimate practical application, and not more than three have

[1] R. H. Thurston, The Steam Engine at the End of the Nineteenth Century, Trans., Am. Soc. M. E., vol. 21, pp. 192 and 228.

stood the commercial test. It would appear that considerable better design must be put in the apparatus if many steps are proposed and also that more consideration must be given to the operating difficulties.

25 Much of the work in connection with the preparation of this paper has been done by W. S. Morrison and M. A. Guigou, members of the Society. Appendices Nos. 1 and 2, on the properties of metals at high temperatures and on the cost of piping materials, are the work of Mr. Morrison; most of the calculations have been done by Mr. Guigou, and the drawings made by Mr. H. H. Worth, Jun. Am. Soc. M. E. Credit should also be given to a number of my friends and associates who have criticised the paper during its preparation.

APPENDIX NO. 1

PROPERTIES OF METALS AT HIGH TEMPERATURES

By W. S. Morrison, New York, N. Y.
Member of the Society

26 Unfortunately we do not have records indicating the service qualities of all of the different metals and their alloys at temperatures much above 700 deg. fahr. For the most part the laboratory experiments so far carried on have not had special reference to particular engineering problems, and were undertaken simply as an attempt to add to our knowledge of the properties of metals.

27 However, we have abundant opportunities for observing the behavior of cast iron and cast steel at temperatures above the normal: the service qualities of cast steel being demonstrated by its use for pipe lines in oil-refining work carrying temperatures above 1000 deg. fahr. Laboratory tests indicate that cast iron does not lose its strength under high temperature for a short period, but that in actual service when subjected to continued temperatures of approximately 450 to 600 deg. fahr. it takes a permanent expansion and does not return to its original volume when cooled. A permanent volumetric expansion is followed by a loss in strength, the loss in cast iron being about 40 per cent in four years.

28 A cast-iron valve 14 in. in diameter is reported to have increased $\frac{5}{16}$ in. in length in four years under an average temperature of 590 deg. fahr.

29 Crane Company made an attempt a short time ago to determine whether cast steel was affected in the same manner as cast iron, and while this experiment was necessarily of short duration the results were quite conclusive for the observed period. Three flanges were taken, one of cast iron, one of cast steel and the third of Ferrosteel. These flanges were subjected to varying degrees of heat for a total period of 130 hours, the maximum temperature reached during that time being 860 deg. fahr. The experiment extended for eighteen days, the flanges being under heat about seven hours each day. The flanges in each case were turned to $12\frac{1}{2}$ in. outside diameter with a bore of $6\frac{29}{64}$ in. It was found that the cast-iron flange increased 0.019 in. in outside diameter and 0.007 in. in inside diameter. The Ferrosteel flange increased 0.033 in. in outside diameter and 0.017 in. in inside diameter. In the case of the cast-steel flange there was no change in dimensions.

30 A great many investigations have been made and reported in the various technical journals, and a few of them seem to have been carried on according to a definite plan. It would appear from a study of the results obtained and the methods employed in the various tests, that the most accurate and important work along these lines has been done since 1920. Among the investigators of this latter period are R. S. MacPherran, of the Allis-Chalmers Company; Dr. S. L. Hoyt; H. J. French, of the

Bureau of Standards; Professor A. K. Huntington, of England; E. L. Dupuy; Prof. F. C. Lea; and the Crane Company, of Chicago.

31 In all of the experiments herein referred to the electric furnace has been chosen for heating the specimens, apparently because of its simplicity of construction, operation, and control. The tensile and elongation tests were for the most part carried out on the usual forms of loaded-beam machines and extensometers, and the temperature readings were taken through the medium of an electric pyrometer, the point of contact being at the center of and on the surface of the test bar. A detailed description of the apparatus employed in each case is given hereinafter under sub-headings indicating the nature of the material under test.

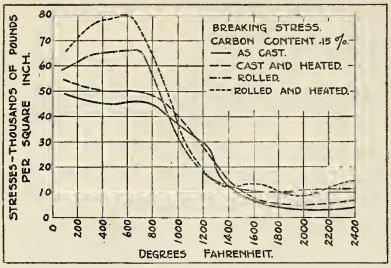
32 *Open-Hearth Steels*—Figs. 3 to 10, inclusive (E. L. Dupuy, *Engineering,* September 9, 1921). In this series of experiments five grades of open-hearth steels were selected, and the investigations had for their object the determination of the properties of these steels in respect to their carbon percentages and were carried out upon both cast and rolled material. These temperature tests were made through the medium of an electric furnace, the metal being protected against oxidation by sheathing the small diameter of the bar in a cylinder of sheet metal resting on the projecting shoulder of the test piece, the intervening space being filled with steel filings or wood charcoal.

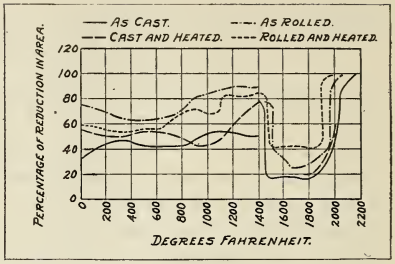
33 The test bars from the cast specimens were cut from the air-cooled ingot without heat treatment of any kind, while the bars from the rolled specimens were cut from a rerolled and reheated billet. The section of the test bars selected for fracture was about 0.233 sq. in. in area, corresponding to the diameter of 0.5434 in. The overall length of the test pieces ranged from 9.055 in. to 11.417 in.; each test piece had a center section 4.331 in. long turned to 0.5434 in. diameter, the two butt ends being turned to a diameter of 0.7874 in.

34 The heat treatment consisted of placing the metal test pieces in a cold furnace and heating them up over a period of $1\frac{1}{2}$ hours to 1697 deg. fahr., which temperature was kept constant for about 10 minutes in order to insure uniformity of temperature between the center and the surface of the bars. This annealing was followed by cooling in still air.

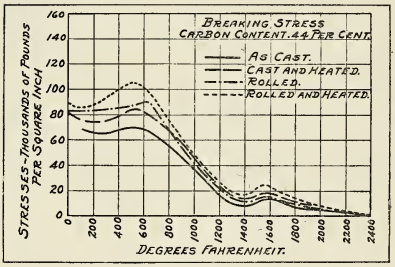
35 The tests were carried out on a 50-ton loaded-beam Falcot machine, the readings being taken on a mercury manometer. The machine was motor-driven and the tests were carried out at a uniform rate of 0.472 in. elongation per minute.

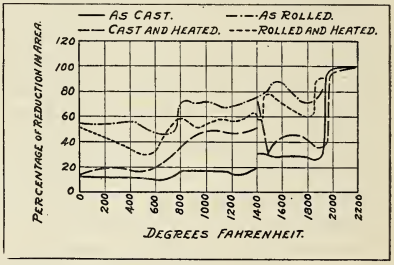
36 In the temperature measurements the hot junction of a platinum-rhodium pyrometer was fixed by means of nichrome wire so as to be in contact with the central portion of the bar. The electromotive force was read by means of a millivoltmeter, enabling the temperature of the thermocouple, and therefore the control of the heating, to be regulated within about 3.6 deg, fahr. The couples were restandardized twice a week. Heating was carried to a temperature 10 deg. higher than that which it was desired to attain, and five minutes more taken in cooling to the final temperature elected. This method of operating secured the equalization of the temperature throughout the bar.



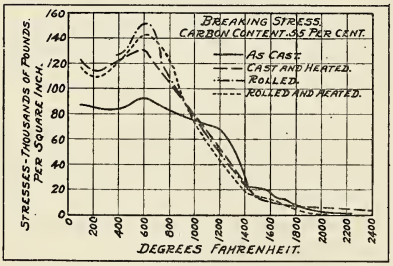


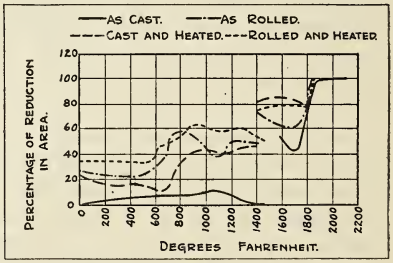
FIGS. 3 AND 4 0.15 PER CENT CARBON STEEL



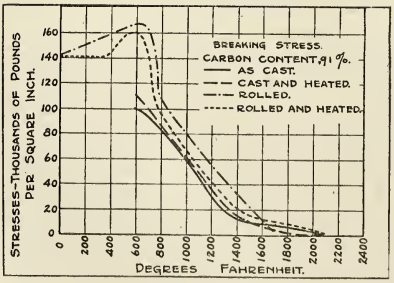


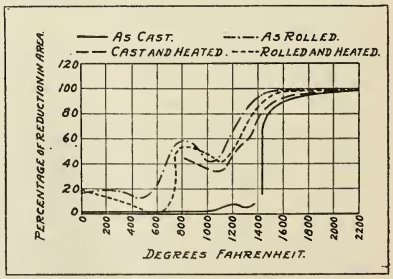
FIGS. 5 AND 6 0.44 PER CENT CARBON STEEL





FIGS. 7 AND 8 0.55 PER CENT CARBON STEEL





FIGS. 9 AND 10 0.91 PER CENT CARBON STEEL

37 The analyses of the steels reported in this investigation are as follows:

	Figs. 3 and 4	Figs. 5 and 6	Figs. 7 and 8	Figs. 9 and 10
Carbon	0.15	0.44	0.55	0.91
Manganese	0.46	0.63	0.70	0.44
Silicon	0.06	0.15	0.52	0.24
Phosphorus	0.014	0.005	0.014	0.019
Sulphur	0.034	0.020	0.023	0.027

38 *Cast Steel*—Fig. 11. The analysis of the metal used in the tests, the results of which are shown, was not obtainable. The data were taken from Marks' Mechanical Engineers' Handbook and it is probably a good grade of steel used for pipe fittings and valves. The same series of tests were reported by Crane Company some time ago.

39 *Pipe Steel*—Fig. 12 (F. N. Speller, Proc. A.S.T.M., Vol. XVI, 1916). The analysis of this steel is: Carbon, 0.15 per cent; manganese, 0.47 per cent; phosphorus, 0.027 per cent; sulphur, 0.27 per cent; silicon, 0.030 per cent.

40 *Boiler Steels*—Figs. 13 and 14 (H. G. French, in *Mining and Metallurgy,* No. 158, Section 15, February, 1920). The steels tested were $\frac{1}{2}$-in. open-hearth boiler plate of firebox and marine grades. The specified tensile strength and the analysis for each are as follows:

Grade	Specified tensile strength, lb. per sq. in.	Carbon, per cent	Manganese, per cent	Phosphorus, per cent	Sulphur, per cent
Firebox	52,000 to 62,000	0.19	0.43	0.020	0.031
Marine	60,000 to 70,000	0.25	0.38	0.019	0.031

41 The test specimens were heated in an electric tube furnace with a heating chamber about 11 in. long. The furnace was operated on either 110 or 220 volts, close regulation being obtained by two 15-ohm, 30-ampere variable resistances in series in the circuit.

42 The apparatus for determining the proportional limit consisted of a frame having an upper and lower half, the specimen being clamped in the center and the frame along with the furnace being assembled between the jaws of the testing machine. The flanges of the upper half of the frame were arranged to receive two Ames dials, the readings of which represented the deformation of the specimen under the load applied.

43 When the desired temperature and thermal equilibrium were reached, a unit load of approximately 1500 to 3000 lb. was applied and the dials set to zero. Readings were then taken at increments of 500 or

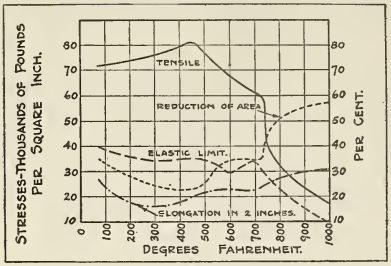


FIG. 11 CAST STEEL

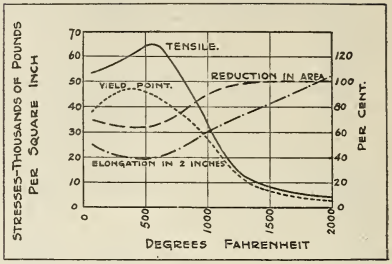


FIG. 12 0.15 PER CENT PIPE STEEL

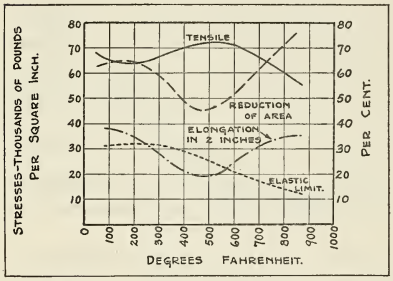


FIG. 13 ½-IN. MARINE BOILER PLATE

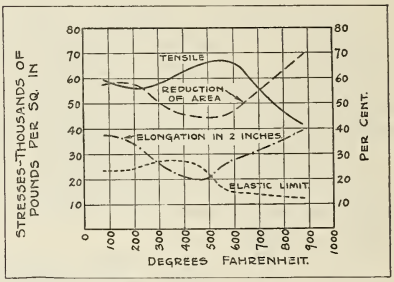
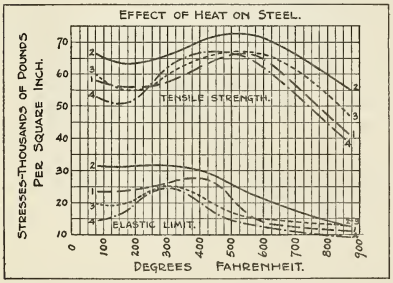
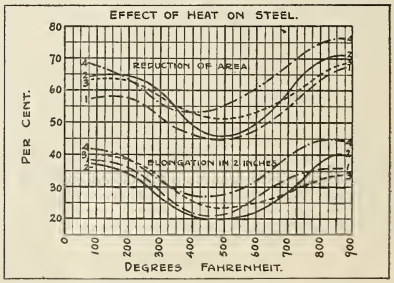


FIG. 14 ½-IN. FIREBOX-GRADE BOILER PLATE





FIGS. 16 AND 17 TANK STEEL

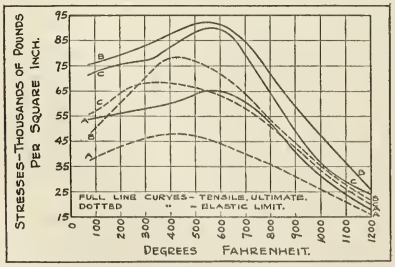


FIG. 15 BOILER STEELS

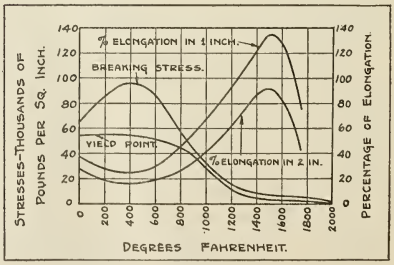


FIG. 18 MILD-STEEL BARS

1000 lb. until the proportional limit had been passed. The dials were then removed and the test piece was broken in the usual manner. Tests at each temperature were made in triplicate and thermal equilibrium was maintained during the actual 8-min. to 15-min. test interval.

44 *Boiler Steels*—Fig. 15 (Lukens Steel Company, Proc. N.E.L.A., 1919). The method of testing employed in this series was not obtainable. The percentage analyses of the specimens are as given in the following table:

	Curve *A*, Fig. 15	Curve *B*, Fig. 15	Curve *C*, Fig. 15
Carbon	0.15	0.210	0.360
Manganese	0.027	0.028	0.029
Phosphorus	0.027	0.018	0.007
Sulphur	0.470	0.490	0.540
Silicon	0.030	0.00	0.005
Nickel	0.00	3.44	0.00

45 *Tank Steel*—Figs. 16 and 17 (The M. W. Kellogg Co.) The analyses of the steels shown in this series of curves are:

	Curve 1, Figs. 16 and 17	Curve 2, Figs. 16 and 17	Curve 3, Figs. 16 and 17	Curve 4, Figs. 16 and 17
Carbon	0.19	0.25	0.17	0.18
Manganese	0.44	0.40	0.34	0.44
Phosphorus	0.021	0.018	0.022	0.015
Sulphur	0.030	0.029	0.032	0.036

46 *Mild-Steel Bars*—Fig. 18 (F. C. Lea, *Engineering,* Aug. 27, 1920). This series of curves shows the results obtained from temperature tests on mild steel which had actually been supplied for concrete reinforcement.

47 *Alloy Steels*—Figs. 19 and 20 (F. C. Lea, *Engineering,* June 30, 1922). The furnace used for heating the test pieces in this series of tests consisted essentially of a steel tube 4 in. in outside diameter and 2 ft. long, wound with nichrome wire surrounded by asbestos, the whole being enclosed in a metal cylinder about 10 in. in diameter provided with steel end plates. Iron-constantin couples were used for measuring the temperature. The specimens, except in the case of the wires, were made 0.564 in. in diameter and the test length was 2 in. The extensometer for determining load strain was set horizontally on a table sliding on steel balls and the relative movement of the tables gave the extension of the specimen.

48 Figs. 19 and 20 have curves numbered 1 to 4, each curve representing a different material, the character and percentage analysis of which are as given below.

Curve 1 — Nickel-chrome steel (Nickel, 3.33 per cent; chromium, 0.61 per cent; vanadium, 0.10 per cent.)

Curve 2 — Chrome-vanadium steel (Chromium, 0.91 per cent; vanadium, 0.13 per cent. Quenched and tempered).

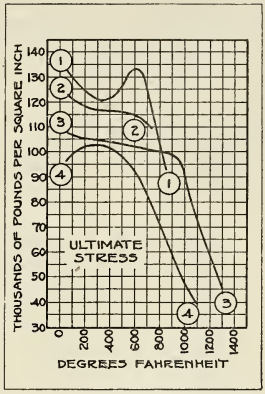


Fig. 19 Alloy Steels

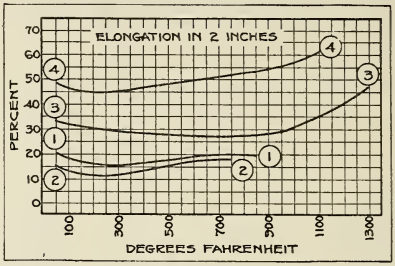


Fig. 20 Alloy Steels

Curve 3 — Nickel-chrome steel wire (Nickel, 60 per cent; chromium, 12 per cent; iron, 24 per cent.)

Curve 4 — 95.5 per cent nickel steel.

49 *Armco Iron* — Fig. 21 (F. C. Lea, *Engineering,* June 30, 1922).

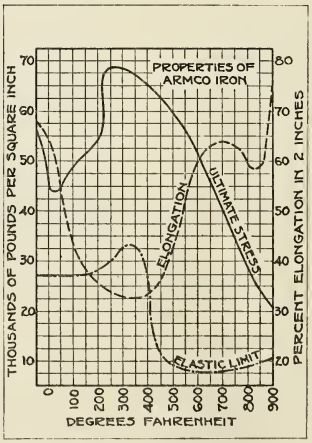


Fig. 21 Armco Iron

This is a comparatively pure wrought iron analyzing about as follows: Carbon, 0.020 per cent; phosphorus, 0.013 per cent; manganese, 0.043 per cent; sulphur, 0.030 per cent.

50 *Monel Metal, Rolled and Cast* — Figs. 22 and 23. Monel metal

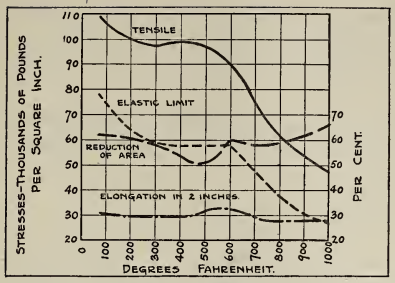


FIG. 22 ROLLED MONEL METAL

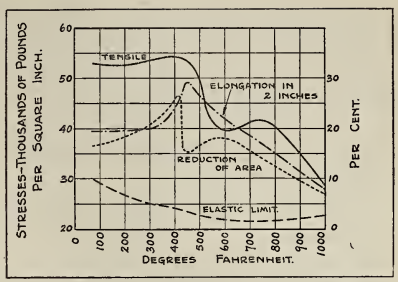


FIG. 23 CAST MONEL METAL

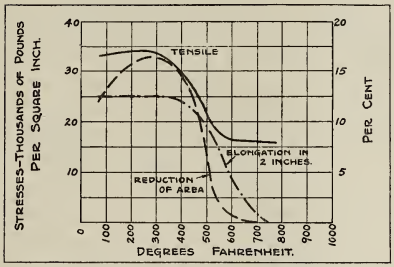


FIG. 24 PHOSPHOR BRONZE

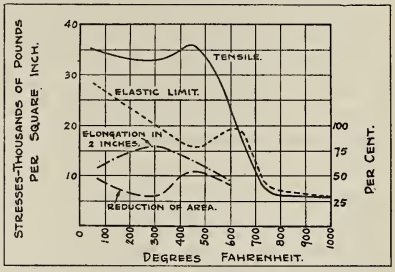


FIG. 25 5 PER CENT ALUMINUM BRONZE

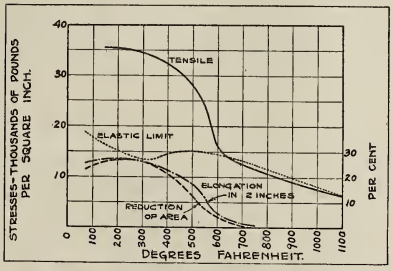


FIG. 26 U.S. NAVY VALVE BRONZE

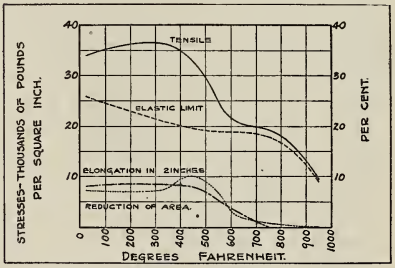


FIG. 27 U.S. NAVY TYPE "G" BRONZE

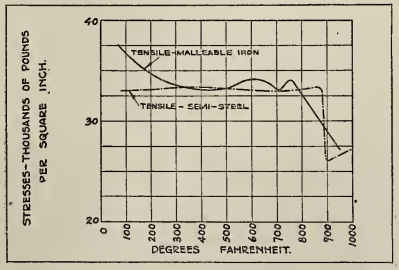


FIG. 28 MALLEABLE IRON AND SEMI-STEEL

is a natural alloy containing 67 to 68 per cent nickel, 28 to 30 per cent copper, and 2 to 5 per cent of other metals, principally iron. The proportions of its various elements are practically the same as those of the ore as it comes from the mine.

51 *Phosphor Bronze* — Fig. 24 (Marks' M. E. Handbook). The analysis quoted is: Copper, 80 per cent; lead, 9 per cent; tin, 10 per cent; phosphorus, 1 per cent.

52 *Five Per Cent Aluminum Bronze* — Fig. 25 (Marks' M. E. Handbook).

53 *U. S. Navy Valve Bronze* — Fig. 26 (Marks' M. E. Handbook). The analysis given is: Copper, 87 per cent; tin, 7 per cent; lead, 1 per cent; zinc, 5 per cent.

54 *U. S. Navy Type "G" Bronze* — Fig. 27 (Crane Company). In this series of tests, as also in tests shown in Fig. 28, the apparatus consisted of a furnace made of a 6-in. sleeve of sheet iron about 6 in. long with $\frac{1}{16}$-in. sheet asbestos for insulation and wound with one layer of nickel resistance wire. One end of the test piece was drilled or cored along its central axis down to within $\frac{1}{2}$ in. of the turned-out section of the specimen and the pyrometer couple was inserted in this opening, which gave the temperature at any time on the inside of the bar very nearly where the fracture was forced to occur. The taking of the elastic limit at slow speed and breaking of the bar, which required about four or five minutes, was accomplished within a degree or two of the desired temperature. The ends of the bar were gripped in the jaws of a Riehlé testing machine, the bar being placed so that the coil was equidistant from the upper and lower jaws and the current turned on until the resistance wire showed the color desired through a piece of mica under a hole in the outer layer of asbestos. The composition of the test pieces of this bronze was 88 per cent copper, 10 per cent tin, and 2 per cent zinc.

55 *Semi-Steel and Malleable Iron* — Fig. 28. The use of semi-steel was introduced into power-plant work about 30 years ago. It is made by melting together about 30 per cent of mild-steel scrap and 70 per cent of pig iron, manganese or other special fluxes being introduced in small amounts to improve the strength and toughness. A metal of about 33,000 to 35,000 lb. tensile strength is produced by this process, but this may be considered only as a good grade of cast iron and for that reason it should not be used in connection with any work where the use of cast iron is prohibited.

56 Malleable iron is midway between cast iron and cast steel in its composition and nature, and for that reason gives a high-strength regular curve.

57 *Open-Hearth and Alloy Steels* — Figs. 29 to 43, incl. (R. S. MacPherran, Proc. A.S.T.M., 1921). These tests were made in the laboratory of the Allis-Chalmers Manufacturing Company to determine the comparative properties of various steels at high temperatures and with a view of obtaining information as to the best material for use under operating conditions of 600 to 1000 deg. fahr. The heating box used was 7 in. square by 9 in. high. The specimens used had threaded ends and were for the most

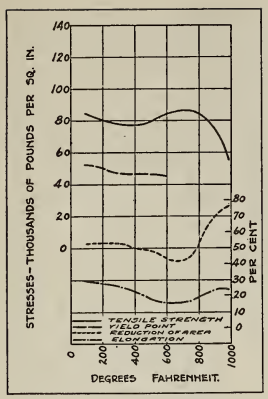


Fig. 29

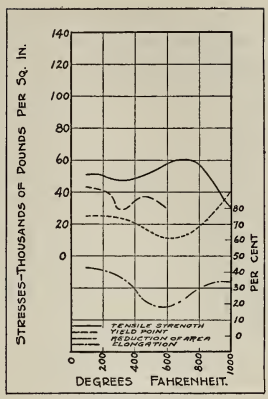


Fig. 30

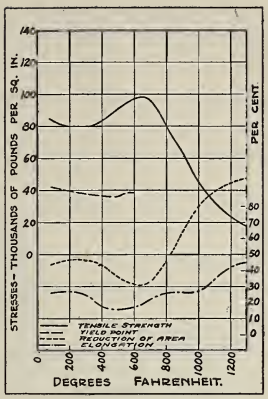


Fig. 31

Figs. 29–31 1-In. Round Rolled Open-Hearth Steel

(Figs. 29 and 30: No heat treatment and unannealed; Fig. 31: annealed at 1475 deg. fahr.

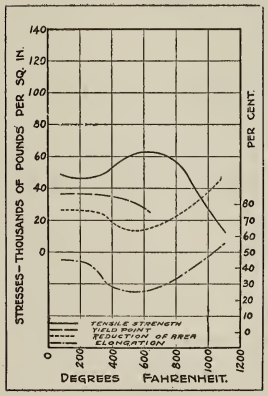


Fig. 32

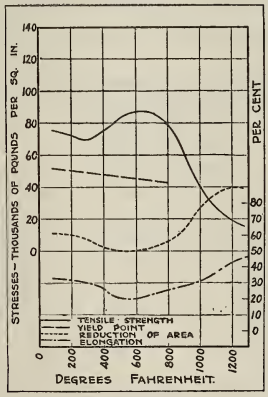


Fig. 33

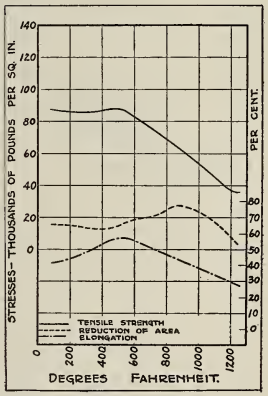


Fig. 34

Fig. 32 $\frac{7}{8}$-In. Round Rolled Open-Hearth Steel

(Annealed at 1475 deg. fahr.)

Fig. 33 Forged 3.25 Per Cent Nickel Steel

(Annealed at 1475 deg. fahr.)

Fig. 34 Forged 34 Per Cent Nickel Steel

part 0.505 in. in diameter. All tests were held at constant temperature for 15 to 30 min. before pulling. The yield points were taken by drop of beam only, and above 600 deg. fahr. are very uncertain.

58 The tungsten-tool-steel specimens were made from one piece of annealed $\frac{7}{8}$-in. round bar about 10 ft. in length, made oversize, then heat treated and ground to definite size.

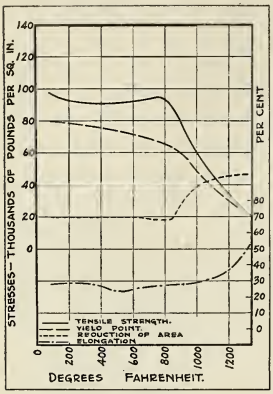


Fig. 35

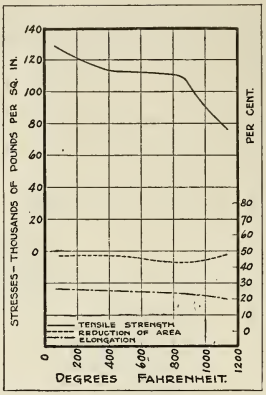


Fig. 36

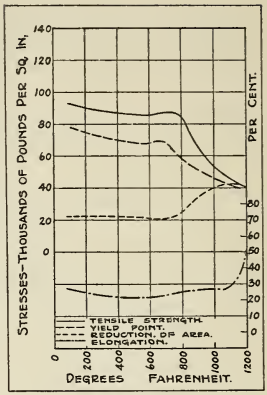


Fig. 37

Fig. 35 Forged Chrome-Nickel Steel

(Quenched from 1500 deg. fahr. in water and drawn at 1300 deg. fahr.)

Fig. 36 Rolled High-Chrome, High-Nickel Steel

(No heat treatment and unannealed.)

Fig. 37 Forged Chrome-Vanadium Steel

(Quenched in water from 1575 deg. fahr. and drawn at 1350 deg. fahr.)

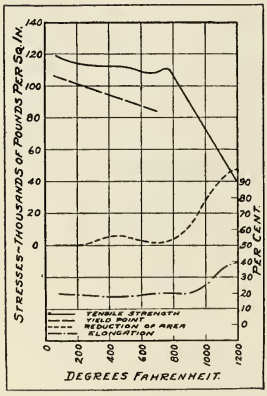


Fig. 38

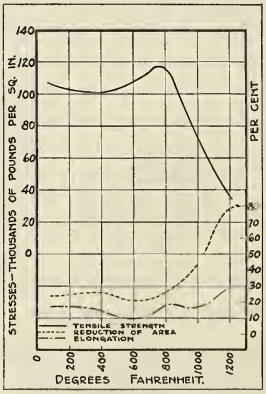


Fig. 39

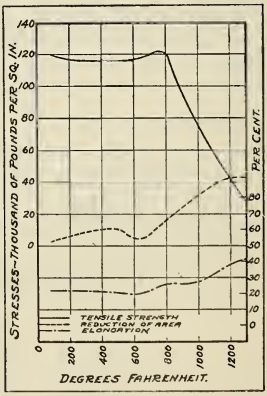


Fig. 40

Fig. 38 Forged High-Chrome Steel

(Quenched from 1550 deg. fahr. and drawn at 1300 deg. fahr.)

Fig. 39 Forged High-Manganese Steel

(Annealed at 1475 deg. fahr.)

Fig. 40 Forged High-Carbon Chrome-Vanadium Steel

(Quenched from 1500 deg. fahr. in water and drawn at 1300 deg. fahr.)

59 In all cases the forged test bars in any one series were made from one billet of steel, this being hammered into several 1-in. rounds about 30 in. in length, and then heat treated in the same furnace.

60 The analyses (percentage) of the metal in the various test pieces are as follows:

1-in. Round Rolled Open-Hearth Steel — Fig. 29: Carbon, 0.41; manganese, 0.59. Not heat treated or annealed.

1-in. Round Rolled Open-Hearth Steel — Fig. 30: Carbon, 0.10; manganese, 0.34. Not heat treated or annealed.

1-in. Round Rolled Open-Hearth Steel — Fig. 31: Carbon, 0.51; manganese, 0.63; phosphorus, 0.01; sulphur, 0.42. Annealed at 1475 deg. fahr.

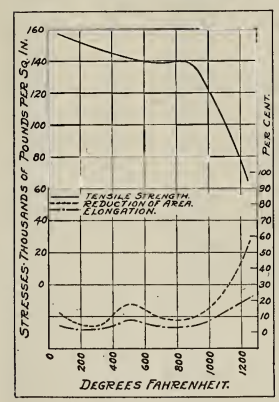


Fig. 41

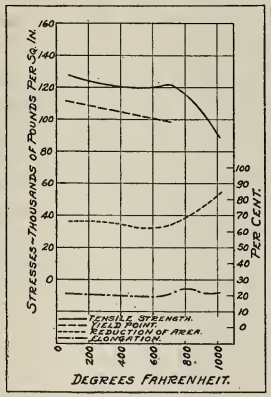


Fig. 42

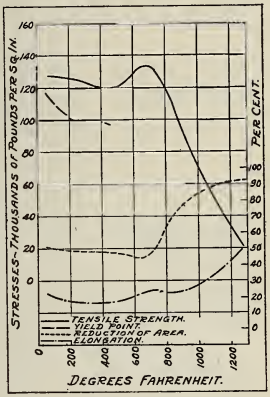


Fig. 43

Fig. 41 Tungsten Tool Steel

(Quenched in oil at 2300 deg. fahr. and drawn at 1400 deg. fahr.)

Fig. 42 Molybdenum Steel

(Quenched in water at 1700 deg. fahr. and drawn at 1300 deg. fahr.)

Fig. 43 High-Manganese, High-Carbon Steel

(Quenched in water at 1475 deg. fahr. and drawn at 1300 deg. fahr.)

$\frac{7}{8}$-in. Round Rolled Open-Hearth Steel — Fig. 32: Carbon, 0.087; manganese, 0.35. Annealed at 1625 deg. fahr.

Forged 3.25 Per Cent Nickel Steel — Fig. 33: Carbon, 0.21; manganese, 0.51; phosphorus, 0.014; sulphur, 0.05; nickel, 3.25. Annealed at 1475 deg. fahr.

Forged 34 Per Cent Nickel Steel — Fig. 34: Carbon, 0.25; manganese, 0.55; phosphorus, 0.013; sulphur, 0.029; silicon, 0.27; nickel, 33.9. Annealed at 1475 deg. fahr.

Forged Chrome-Nickel Steel — Fig. 35: Carbon, 0.34; manganese, 0.59; phosphorus, 0.013; sulphur, 0.012; silicon, 0.10; chromium, 0.38; nickel, 2.38. Quenched from 1500 deg. fahr. in water and drawn at 1300 deg. fahr.

TABLE 4 APPROXIMATE TENSILE STRENGTH OF MATERIALS TESTED AT 70 DEG. FAHR. AND AT THE TEMPERATURE OBSERVED AT MAXIMUM TENSILE STRENGTH

Investigator	Fig. no.	Material tested	Approx. tensile strength at 70 deg. fahr. lb. per sq. in.	Observed temp. at max. tensile strength, deg. fahr.	Approx. max. tensile strength at observed temp., lb. per sq. in.
E. L. Dupuy	3	0.15% carbon steel as cast	50,000	700	50,000
	3	0.15% carbon steel cast and heated	52,000	700	52,000
	3	0.15% carbon steel rolled	59,000	700	67,000
	3	0.75% carbon steel rolled and heated	65,000	600	80,000
	5	0.44% carbon steel as cast	68,000	500	70,000
	5	0.44% carbon steel cast and heated	66,000	550	85,000
	5	0.44% carbon steel rolled	84,000	625	91,000
	5	0.44% carbon steel rolled and heated	85,000	450	106,000
	7	0.55% carbon steel as cast	87,000	600	93,000
	7	0.55% carbon steel cast and heated	121,000	600	130,000
	7	0.55% carbon steel rolled	.....	600	151,000
	7	0.55% carbon steel rolled and heated	116,000	600	142,000
	9	0.91% carbon steel as cast	.....	600	100,000
	9	0.91% carbon steel cast and heated	.....	600	110,000
	9	0.91% carbon steel rolled	146,000	600	167,000
	9	0.91% carbon steel rolled and heated	140,000	600	160,000
Crane Company	11	Commercial cast steel	72,000	450	84,000
F. N. Speller	12	0.15% carbon pipe steel	57,000	600	66,000
H. G. French	13	0.25% carbon marine-grade boiler steel	68,000	550	73,000
	14	0.19% carbon firebox boiler steel	60,000	550	67,000
Lukens Steel Co.	15	0.15% carbon boiler steel	58,000	550	65,000
		0.21% carbon boiler steel	75,000	550	94,000
		0.36% carbon boiler steel	71,000	550	90,000
M. W. Kellogg Co.	16	0.19% carbon tank steel	57,500	525	67,000
		0.25% carbon tank steel	65,250	525	72,500
		0.17% carbon tank steel	60,000	525	67,000
		0.18% carbon tank steel	50,250	450	67,000
F. C. Lea	18	Mild-steel bars	63,000	400	88,000
	19	Nickel-chrome steel (nickel 3.33%)	130,250	600	133,000
		Chrome-vanadium steel (chrome 0.91%)	123,000	70	123,000
		Nickel-chrome steel wire (nickel 60%)	100,750	70	100,750
		95.5% nickel steel	96,000	250	103,000
Internat'l Nickel Company	21	Armco iron	57,000	275	68,000
	22	Rolled monel metal	108,000	400	99,000
	23	Cast monel metal	53,000	400	54,500
	24	Phosphor bronze (80% copper)	33,000	300	34,000
	25	5% aluminum bronze	35,000	450	36,000
Crane Company	26	U. S. Navy valve bronze (87% copper)	34,000	70	34,000
	27	U. S. Navy Type "G" bronze (88% copper)	34,000	300	36,500
	28	Malleable iron	37,500	70	37,500
	28	Semi-steel	33,000	400	33,500
R. S. MacPherran	29	0.41% carbon rolled steel, untreated	85,000	700	87,500
	30	0.10% carbon rolled steel, untreated	52,000	700	60,000
	31	0.51% carbon rolled steel, annealed	82,500	680	98,000
	32	0.087% carbon rolled steel, annealed	49,500	625	62,000
	33	3.25% nickel steel, annealed	74,000	625	87,000
	34	34% nickel steel, annealed	87,000	500	88,000
	35	0.38% chrome-nickel steel, quenched and drawn	98,000	70	98,000
	36	High-chrome, high-nickel steel, untreated	128,000	70	128,000
	37	Chrome-vanadium steel, quenched and drawn	93,000	70	93,000
	38	High-chrome steel, quenched and drawn	119,000	70	119,000
	39	High-manganese steel, annealed	107,000	780	117,000
	40	High-carbon chrome-vanadium steel, quenched and drawn	121,000	760	122,000
	41	19.28% tungsten tool steel, quenched and drawn	157,000	70	157,000
	42	Molybdenum steel, quenched and drawn	128,000	70	128,000
	43	High-manganese, high-carbon steel, quenched and drawn	128,000	680	133,000

Rolled High-Chrome, High-Nickel Steel — Fig. 36: Carbon, 0.53; manganese, 0.58; silicon, 1.25; chromium, 7.42; nickel, 20.78; iron, 69.34. Not heated or annealed.

Forged Chrome-Vanadium Steel — Fig. 37: Carbon, 0.30; manganese, 0.52; phosphorus, 0.010; sulphur, 0.031; silicon, 0.18; chromium, 0.83; vanadium, 0.17. Quenched in water from 1575 deg. fahr. and drawn at 1300 deg. fahr.

Forged High-Chrome Steel — Fig. 38: Carbon, 0.69; manganese, 0.27; phosphorus, 0.015; sulphur, 0.017; silicon, 0.18; chromium, 3.36. Quenched from 1550 deg, fahr. in water and drawn at 1300 deg fahr.

Forged High-Manganese Steel — Fig. 39: Carbon, 0.72; manganese, 0.92; phosphorus, 0.016; sulphur, 0.38; silicon, 0.110. Annealed at 1475 deg. fahr.

Forged High-Carbon Chrome-Vanadium Steel — Fig. 40: Carbon, 0.82; manganese, 0.38; phosphorus, 0.01; sulphur, 0.032; silicon, 0.41; chromium, 0.91; vanadium, 0.18. Quenched from 1500 deg. fahr. in water and drawn at 1300 deg. fahr.

Tungsten Tool Steel — Fig. 41: Carbon, 0.68; manganese, 0.12; phosphorus, none; sulphur, none; silicon, 0.23; tungsten, 19.28; chromium, 3.36; vanadium, 0.88; molybdenum, trace. Quenched in oil at 2300 deg. fahr. and drawn at 1400 deg. fahr.

Molybdenum Steel — Fig. 42: Carbon, 0.38; manganese, 0.62; phosphorus, 0.011; sulphur, 0.014; silicon, 0.15; chromium, 0.86; molybdenum, 0.34. Quenched in water at 1700 deg. fahr. and drawn at 1300 deg. fahr.

High-Manganese, High-Carbon Steel — Fig. 43: Carbon, 0.75; manganese, 1.04; phosphorus, 0.033; sulphur, 0.042; silicon, 0.07. Quenched in water at 1475 deg. fahr. and drawn at 1300 deg. fahr.

61 Table 4 gives the approximate tensile strength at room temperature, the temperature observed at maximum tensile strength, and the approximate maximum tensile strength at that temperature for the entire series of curves.

62 Space does not permit of a discussion of the results obtained from the tests herein described, but it may be said that our present specifications for materials are based upon experience under conditions quite different from those existing in many of the large power stations at the present time, and it is conceivable that much better results may be obtained with materials purchased under very different specifications. For example, a higher carbon content in steel boiler tubes would probably increase the strength in approximately the same proportion at all temperatures, the main trouble being that at temperatures of from 900 to 1000 deg. fahr. the tube starts to scale by decomposition. Research work is now under way to determine a method for overcoming this natural tendency of the material.

63 In the foregoing series of experiments the time consumed during test was so short as to give very little idea of the probable behavior of the metal when maintained for considerable periods under the conditions of stress and temperature existing in the test specimen.

64 Some recent experimental work by J. H. S. Dickenson (Sheffield,

TABLE 5 HEAT TREATMENT AND ANALYSES OF STEELS USED IN TESTS REFERRED TO IN TABLE 6

Class of Steel	Diam. of Bar. in.	Heat Treatment before Machining Test Piece	Analyses								
			Carbon	Silicon	Manganese	Sulphur	Phosphorus	Nickel	Chromium	Tungsten	Vanadium
0.30 % carbon....	$\frac{7}{8}$	Oil-hardened at 1562 deg. fahr., Tempered at 1067 deg. fahr.	0.30	0.12	0.54	0.027	0.038	0.51	0.14	.....	.....
0.45 % carbon....	1	Oil-hardened at 1562 deg. fahr., Tempered at 1112 deg. fahr.	0.45	0.37	0.79	0.022	0.024	.....	0.02	.....	.....
Nickel-chrome...	$\frac{3}{4}$	Oil-hardened at 1526 deg. fahr., Tempered at 1112 deg. fahr.	0.25	0.11	0.36	0.031	0 026	3.63	0.55	.....	.....
High-chrome....	$\frac{7}{8}$	Oil-hardened at 1697 deg. fahr., Tempered at 1202 deg. fahr.	0.26	0.59	0.17	0.066	0.016	0.39	14.68	.....	.....
High-speed.....	$\frac{13}{16}$	Annealed at 1472 deg. fahr.	0.60	0.28	0.23	0.044	0.029	.....	3.49	14.09	0.72
Nickel-chrome alloy, cast.....	.....	.....	0.54	0.73	0.10	0.032	0.030	69.90	15.50	.....	.....

TABLE 6 BEHAVIOR OF VARIOUS STEELS WHEN SUBJECTED TO TENSILE STRESS FOR CONSIDERABLE PERIODS AND AT TEMPERATURES RANGING FROM 932 DEG. FAHR. TO 1769 DEG. FAHR.

(1) Class of Steel	(2) Temperature Range, deg. fahr.	(3) Duration of Test in Hours before Fracture or Withdrawal	(4) Time at Load of 19000 lb. per sq. in. and Temperature as stated in. Col. (2)	(5) Elongation on middle 2 in., per cent	(6) Reduction of area per cent
0.30% carbon	932–1022	1739–withdrawn	3400 hr. [1]	.....	.....
	1022–1112	956–broken	956 hr.	30.0	85
	1112–1202	28–broken	28 hr.	29.0	88
	1292	.....	3 min.	32.5	92
	1427	.....	6 sec. [1]	.....	.....
0.45% carbon	932–1022	1405–withdrawn	2600 hr. [1]	.....	.....
	1022–1112	195–broken	195 hr.	32.0	84
	1112–1202	26–broken	26 hr.	32.0	84
	1256	.....	3 min.	33.0	90
	1427	.....	6 sec. [1]	.....	.....
Nickel-chrome	932–1022	701–broken	701 hr.	40.0	85
	1022–1112	88–broken	88 hr.	41.0	88
	1112–1202	8–broken	8 hr.	38.0	91
	1265	.....	3 min.	40.0	94
	1481	.....	6 sec. [1]	.....	.....
14.7% chrome	932–1022	1740–withdrawn	10000 hr. [1]	.....	.....
	1022–1112	1728–broken	1728 hr.	34.0	84
	1112–1202	79–broken	79 hr.	37.0	85
	1355	.....	3 min.	36.0	91
	1472	.....	6 sec. [1]	.....	.....
High-speed	932–1022	1717–withdrawn	81000 hr. [1]	.....	.....
	1022–1112	2002–withdrawn	25000 hr. [1]	.....	.....
	1112–1202	380–withdrawn	4500 hr. [1]	.....	.....
	1202–1292	564–broken	564 hr.	19.0	47
	1292–1382	21–broken	21 hr.	24.0	60
	1499	.....	3 min.	32.0	82
	1751	.....	6 sec. [1]	.....	.....
Nickel-chrome alloy (as cast)	932–1022	521–withdrawn	.....	.....	.....
	1022–1112	3604–withdrawn	108120 hr. [1]	.....	.....
	1112–1202	6041–broken	6041 hr	17.5	40
	1202–1292	901–broken	901 hr.	14.5	23
	1292–1382	273–broken	273 hr.	27.0	51
	1643	.....	3 min.	24.5	44
	1697	.....	1 min.	25.0	47
	1769	.....	6 sec. [1]	.....	.....

[1] Estimated

England) reported in *Engineering* for September 15, 1922, calls attention to the influence of the time factor in determining the temperature up to which certain steels can support a given load, and, by implication, the load which can be borne at any given temperature.

65 Two series of tests were made, one to determine duration at constant load and constant temperature, the other at constant load and a uniform rate of temperature increase. In both tests the load on the specimen was 19,000 lb. per sq. in. and the temperature varied from 932 deg. fahr. to 1769 deg. fahr.; and the tests were carried out on six samples of steel made into specimen bars 8 in. long and 0.40 in. in diameter gage length. All of the steels, with the exception of the alloy of cast nickel-chrome, were heat-treated. The classes of steels used, their analyses, and heat treatments are given in Table 5.

66 The apparatus used was arranged for six tensile tests simultaneously with the electric furnaces wired in parallel, temperatures being recorded by a Cambridge thread recorder, the couples being checked before and after use.

67 *Duration Tests at Constant Load and Constant Temperature.* Care was taken in applying the load so that no live load was momentarily produced and the temperature was steadily raised to the desired point, where it was maintained day and night until fracture occurred or until it was deemed unnecessary to continue; a daily measurement being made of the change of distance between the gage points on the specimen. More frequent observations were made when called for by rapid extension or when rupture became imminent at the end of long runs.

68 A series of test pieces from each steel was run in 90-deg. ranges, as from 1022 to 1112 deg. fahr., 1112 to 1202 deg. fahr., and so on. Where the temperature range for any given specimen is stated as, say, from 1112 deg. fahr. to 1202 deg. fahr., the mean temperature was not far from 1157 deg. fahr., especially in the longer runs.

69 *Tests at Constant Load and Uniform Rate of Temperature Rise.* After the test piece was placed in position and the load applied the gage length was measured. The temperature was then slowly raised by means of a specially devised rheostat at the uniform rate of 212 deg. fahr. in every 60 min., an overall measurement being made at 212 deg. fahr. and thereafter on reaching each additional 90 deg. fahr., care being taken to make this coincide with half-hourly periods. Later as the rate of extension increased more frequent measurements were made, usually in 6-min. periods. Finally the temperature at which the test piece ruptured was carefully noted. Table 6 gives the test results for the entire range of the two series of experiments. It also includes figures extracted from graphs included in the original article, at which the four steels, 0.30 carbon, 0.45 carbon, nickel-chrome and high-chrome, would rupture under a load of 19,000 lb. per sq. in. under the loading conditions described for the rapid tests. In these cases the duration full load is estimated at 6 sec. It is necessary also to state, in the same table, the period during which the test piece of the series "constant load and rising temperature," were at the temperature of breakdown. The figure given is 3 min., the time occupied in rising the 50 deg. fahr. prior to final collapse of the test piece. The remaining estimates in Table 6 are those of the probable lives of

test pieces subjected to constant load and temperature for long periods and eventually withdrawn, stretched but not broken. An estimate of the probable time before rupture would have occurred, had the test proceeded, was obtained in each case by comparing the time required to produce the same extension

TABLE 7 TEMPERATURES UNDER DIFFERENT LOAD CONDITIONS FOR STEELS SUPPORTING A LOAD OF 19.000 LB. PER SQ. IN.

Class	Under a rapidly applied load or for short duration of loading	When the specimen is expected to endure for considerable periods without suffering sensible deformation
0.30 per cent carbon steel ...	1357 deg. fahr.	932 deg. fahr.
Nickel-chrome steel.....	1481 deg. fahr.	842 deg. fahr.
High-chrome steel.....	1472 deg. fahr.	939 deg. fahr.
High-speed steel.....	1751 deg. fahr.	1067 deg. fahr.
Nickel-chrome alloy (cast)..	1767 deg. fahr.	1112 deg. fahr.

in unbroken and broken specimens of the same steel. These estimates, however, are only of general interest.

70 The general deduction obtained from Table 6 is that the temperature attained is higher as the duration at load is shorter and that the

TABLE 8 ANALYSES OF STEELS USED IN EXPERIMENTS TO DETERMINE SCALING RATES AT ELEVATED TEMPERATURES

Class of Steel	Analyses							Heat Treatment
	Carbon	Silicon	Manganese	Sulphur	Phosphorus	Nickel	Chromium	
0.30% carbon.....	0.30	0.12	0.54	0.027	0.038	0.51	0.014	Oil-treated
0.45% carbon.....	0.45	0.37	0.79	0.022	0.024	.....	0.02	Oil-treated
Nickel chrome.....	0.28	0.20	0.31	0.034	0.033	3.70	0.71	Oil-treated
5% nickel.....	0.10	0.06	1.12	0.032	0.036	5.43	0.14	Normalized
25% nickel.....	0.27	0.16	0.21	0.022	0.024	24.36	0.06	Water-quenched
High-speed [1].....	0.85	0.40	0.23	0.044	0.022	0.08	4.19	Annealed
High-chrome.....	0.26	0.59	0.17	0.066	0.016	0.39	14.68	Oil-treated
Nickel-chrome alloy...	0.29	0.87	1.80	0.050	0.018	65.05	11.70	Water-quenched

[1] This steel also contains 18.36 per cent tungsten and 0.96 per cent vanadium

reduction of area rises as the test pieces are more rapidly broken, and that this is some times accompanied by a corresponding rise in the percentage of elongation. The three high-speed steel specimens, broken in 564 hours, 21 hours, and 3 min. (at load) provide the most striking instance of this combined increase.

71 Table 7 gives the probable range of maximum temperature up to which each of the typical steels will support a load of 19,000 lb. per sq. in. with the given conditions of loading.

72 An extension of the article above referred to, appearing in *Engineering* for September 22, 1922, deals with the scaling of steels at elevated temperatures. Eight typical steels, five of which were those given in Table 5, were selected for examination. From each sample of steel, nine cylinders, each 0.50 in. in diameter by 2 in. long were machined, polished with emery, and weighed. Each of these test cylinders was heated for a total of 100 hours. In order to maintain throughout the 100 hours a practically uniform rate of oxidation — which slows down as adhering scale increases in thickness — the heating was carried out in 18 periods of approximately 5½ hours each; the specimens being scraped free of scale and weighed after each cooling. Two

TABLE 9 SCALING RATES OF VARIOUS STEELS AT TEMPERATURES RANGING FROM 1067 DEG. FAHR. TO 2012 DEG. FAHR.

Class of Steel	Scaling Rate in Ounces per Square Inch at Mean Temperature Indicated								
	Heated in Electric Furnace to					Heated in Gas Furnace to			
	1067 deg. fahr.	1247 deg. fahr.	1337 deg. fahr.	1427 deg. fahr.	1607 deg. fahr.	1607 deg. fahr.	1733 deg. fahr.	1877 deg. fahr.	2012 deg. fahr.
0.30% carbon	0.0027	0.0080	0.018	0.036	0.100	0.094	0.160	0.37	0.55
Nickel-chrome	0.0027	0.0095	0.014	0.025	0.080	0.070	0.130	0.32	0.60
25% nickel	0.0005	0.0022	0.005	0.011	0.054	0.023	0.100	0.21	0.41
High-speed	0.0011	0.0038	0.006	0.010	0.054	0.060	0.095	0.25	0.47
Nickel-chrome alloy	nil	nil	....	....	0.0017	0.0011	0.0014	0.002	0.008
High-chrome	nil	0.0005	0.001	0.002	0.007	0.008	0.017	0.14	0.27

types of furnaces were used for heating the specimens, one electric and one gas-heated furnace.

73 The two carbon steels showed almost exactly the same scaling rate throughout, as did also the two high-speed steels. A 5 per cent nickel steel and the nickel-chrome steel also behaved so much alike that only the results from the former are here given. Table 8 gives the analyses of the steels used in these tests and Table 9 the scaling rates in ounces per square inch per hour at the mean temperatures indicated, the latter table including the results derived from both the electric and the gas furnace. The lower-temperature series were heated in an electric furnace where the specimens were surrounded by air and the higher ranges heated in the less pure atmosphere of the gas muffle. It appears that the rate of scaling at about 1600 deg. fahr. is much the same in the two types of furnace, at any rate when the gas-muffle front is slightly open and the burners are receiving full air, as in the present case, so that the lower and upper may thus be considered satisfactorily linked.

74 Plotting the scaling rates in ounces per square inch against temperature shows at a glance the temperature at which scaling becomes appreciable on each of the steels examined. If, for example, a scaling rate of 0.00018 oz. per sq. in. per hour was to be fixed as the maximum allowable, the nickel-chrome and carbon steel could not be used at over 1148 deg. fahr., high-speed steel could go up to 1310 deg. fahr., high-chrome steel up to 1544 deg. fahr., and the nickel-chrome alloy up to no less than 1958 deg.

75 These tests of Mr. Dickenson tend to cast grave doubt on our ideas as to the yield point and elastic limit of certain of the steels in common use. It would appear that we may be compelled to revise our ideas along these particular lines, and as it is now generally agreed that all design work should be based either on the yield point or the elastic limit, this matter becomes important.

APPENDIX NO. 2

COST OF PIPING MATERIALS

By W. S. Morrison, New York, N. Y.
Member of the Society

76 Figs. 44 to 50, inclusive, show that the unit cost of pipe fittings increases in a certain proportion with the increase in diameter of the fitting — in the case of the tees and elbows at least. The curves as drawn represent averages of prices recently obtained from manufacturers. The cast-iron and cast-steel fittings characterized as "low pressure," "Standard" and "extra heavy" are of the type and body thickness required under their respective standards. The curves in Fig. 47 represent fittings for 350 lb. pressure which are known as "hydraulic fittings," and are usually guaranteed for a temperature of 800 deg. fahr.

77. The curves in Fig. 48 give the percentage of increase in price with increase of pressure for cast-iron valves with bronze trimmings, the unit cost at 100 lb. pressure being taken as a basis.

78 Fig. 49 shows the average unit price of cast-steel, monel-metal-trimmed valves for a pressure of 350 lb. and 800 deg. fahr. total temperature, and Fig. 50 the percentage increase in price for the same type of valve plotted against diameters, for pressures of 400 and 500 lb. per sq. in.

79 The probable increase in cost of a complete plant system of piping, including pipe, pipe fittings, valves, covering, etc., for both steam and water within the range of from 200 lb. pressure to 500 lb. pressure, based on current prices for material, is fairly represented by the curve of Fig. 51, taking the piping cost at 200 lb. pressure as a basis.

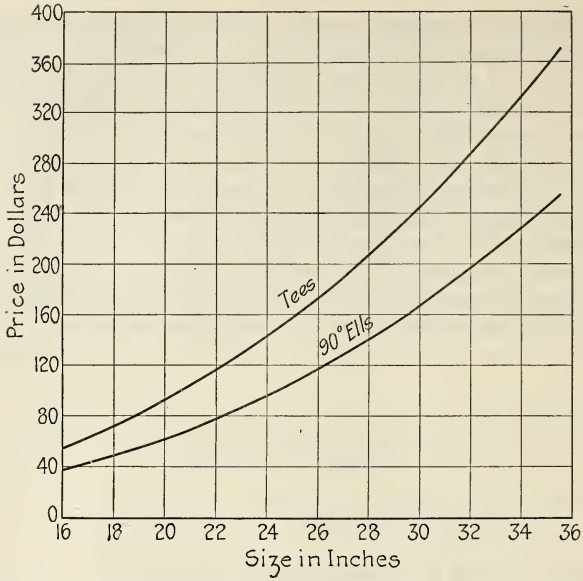


Fig. 44 "Low-Pressure" Cast-Iron Fittings — Size vs. Price

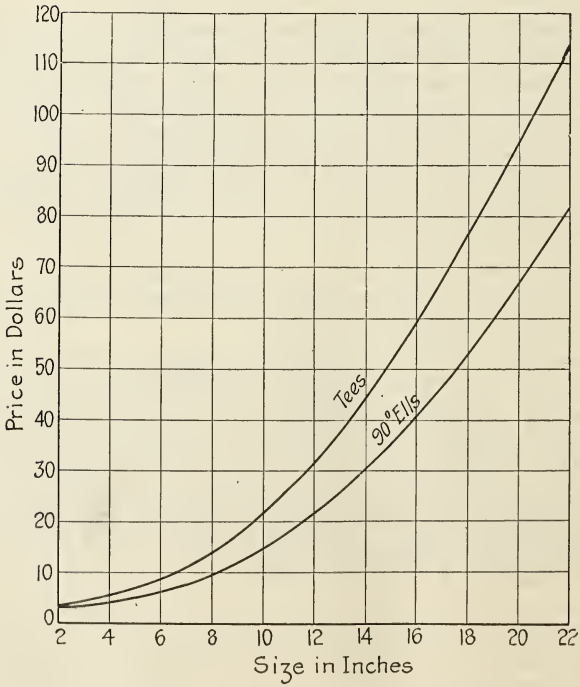


Fig. 45 "Standard" Cast-Iron Fittings — Size vs. Price

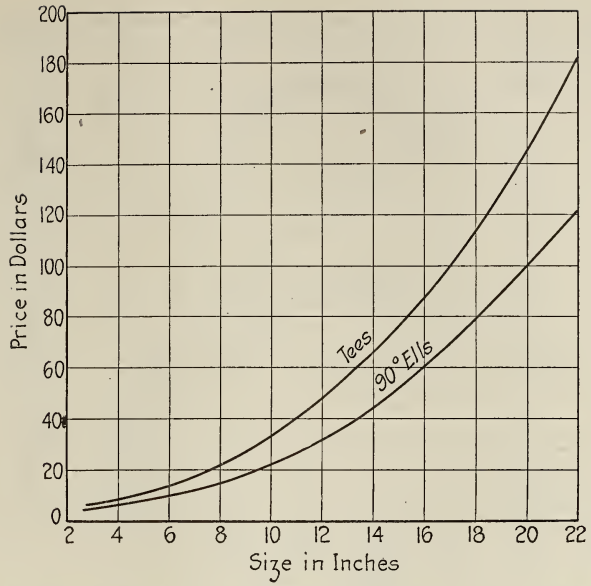


Fig. 46 "Extra Heavy" Cast-Iron Fittings (250 lb. Pressure) — Size vs. Price

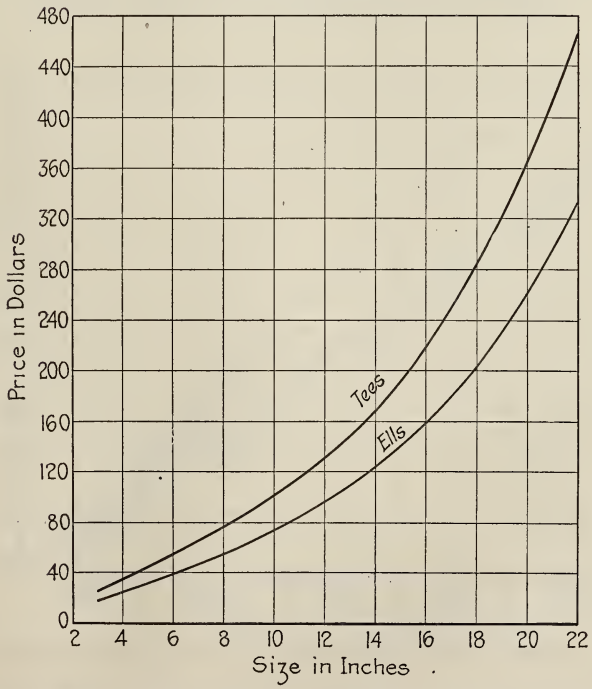


Fig. 47 Commercial "Extra Heavy" Cast-Steel Fittings (350 lb. Pressure) — Size vs. Price

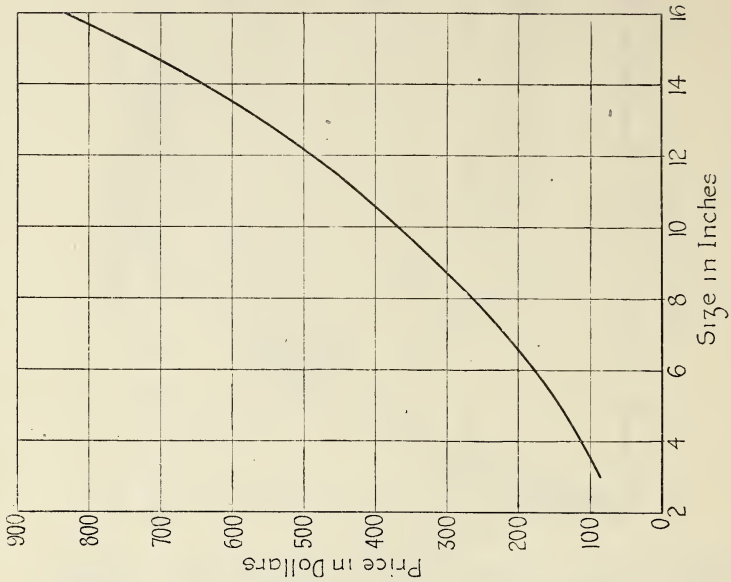


Fig. 49 Average Unit Prices of "Extra Heavy" Cast-Steel Gate Valves (Monel-Metal Trim; for 350 lb. Pressure and 800 Deg. Fahr.)

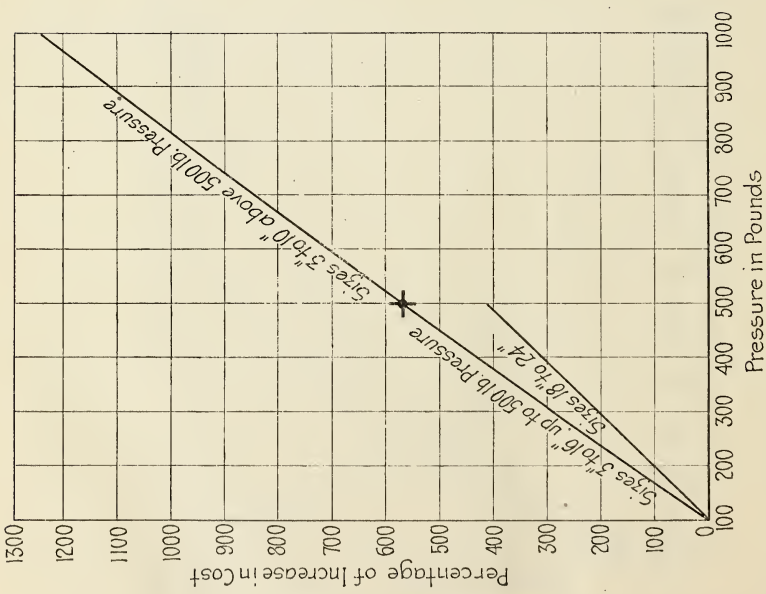


Fig. 48 Curves Showing Percentage of Increase in Cost of Cast-Iron Valves (O.S.&Y.) with Rise in Pressure, Cost at 100 lb. Pressure as Basis

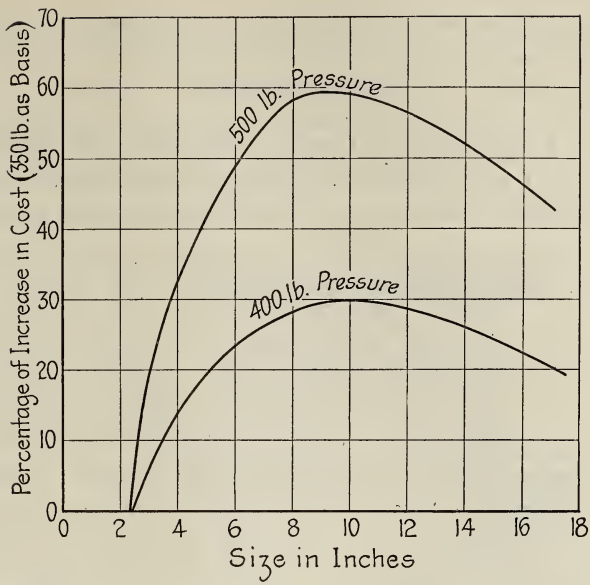


Fig. 50 Curves Showing Percentage of Increase in Cost of Cast-Steel Valves (O.S.&Y.) with Rise in Pressure, Cost at 350 lb. Pressure as Basis (Maximum Temperature, 800 Deg. Fahr.)

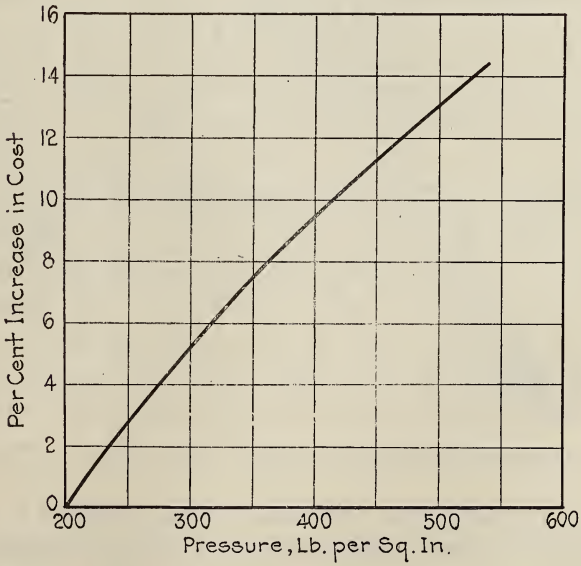


Fig. 51 Percentage of Increase in Piping Cost with Increase in Pressure

DISCUSSION

John A. Stevens and Carl J. Sittinger. Mr. Orrok's paper indicates that we have not yet reached the ultimate in economy of steam plants. In fact, it is possible and indeed probable that

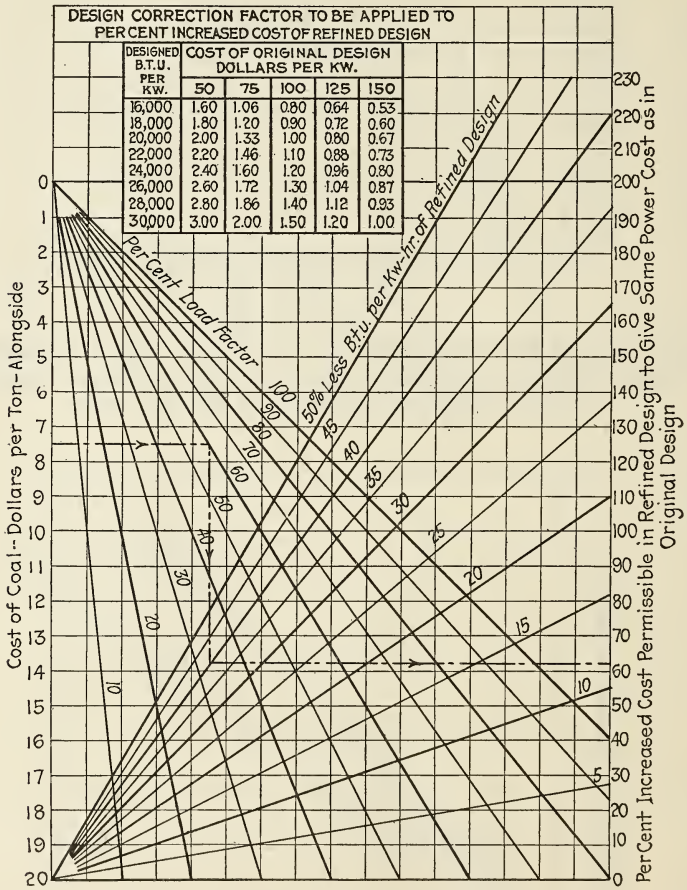


Fig. 52 Relation Between Coal Cost, Load Factor and Plant Costs

there will be, during the next decade or so, as great a percentage reduction in the heat consumption per kw-hr. as there has been in the last fifteen years. Just how far it will be possible to effect

this reduction depends, as Mr. Orrok has clearly brought out, on how much the necessary refinements will cost and whether the saving in operation will offset the additional fixed charges. Fuel cost and load factor, of course, have predominating influences on how far it is possible to carry out such refinements.

The writers would like, at this time, to present another method of attacking this problem. It is possible to show mathematically a relation between coal cost, load factor, and plant costs, and with slight approximation this relation reduces to a very simple form with an error within 2 per cent.

Fig. 52 is a graphical representation of this relation by which one can easily decide how far refinements can be carried without increasing the net cost of power.

Let us assume two initial designs, — one at $100 per kw. for a 20,000-B.t.u. station giving a correction factor of 1.00, and another at $125, per kw. for an 18,000-B.t.u. station giving a correction factor of 0.72. With coal at $7.50 a ton and 60 per cent load factor, how much could we afford to pay for a station giving an economy of 12,000 B.t.u. per kw-hr.

For the first case, the 12,000-B.t.u. refined design means 40 per cent less B.t.u. per kw-hr., for which it would be permissible to pay 62 per cent $\times$ 1.00 correction factor = 62 per cent more than the initial design, or $162 per kw.

For the second case, the 12,000-B.t.u. refined design means $33\frac{1}{3}$ per cent reduction in B.t.u. per kw-hr. for which it would be permissible to pay 51 per cent $\times$ 0.72 correction factor = 36.7 per cent more than the initial design, or $171.

In Table 10 is a tabulation of the possibility of two refined designs at 40 per cent and 60 per cent load factor, and at various prices of coal from $2.00 to $20.00 per ton. These were taken directly from the curves shown in Fig. 52.

TABLE 10 INFLUENCE OF COAL COST AND LOAD FACTOR ON EFFICIENCY AND FIRST COST OF POWER PLANT

Design Correction Factor, — 1.00
Original Design Assumed to Cost $100 per kw. With an Efficiency of 20,000 B. per kw. hr.

Cost of Coal Per Ton Dollars	Per Cent Increased Cost Permissible in Refined Design			
	15,000 B.T.U. or 25 per cent less		12,000 B.T.U. or 40 per cent less	
	Load Factor, per cent		Load Factor, per cent	
	40	60	40	60
2.00	6.9	10.3	11.0	16.5
3.00	10.3	15.5	16.5	24.8
4.00	13.8	20.7	22.0	33.0
5.00	17.2	25.8	27.5	41.3
6.00	20.7	31.0	31.1	49.6
7.50	25.9	38.8	41.3	62.0
10.00	34.5	51.7	55.1	82.6
15.00	51.7	77.5	82.9	124.0
20.00	69.0	103.3	110.2	165.2

If original design had cost $125 per kw. with an efficiency of 18,000 B.t.u. per kw-hr., the design correction factor would have been 0.72 and the above figures would have been 28 per cent less.

I. E. Moultrop. Mr. Orrok has pointed out very definitely the present limits of temperature which station designers must meet. The writer believes the author's statements in reference to the possibilities of making boilers for the higher pressures is quite reasonable but he cannot feel so optimistic about the possibilities of producing piping, valves and fittings for such pressures and temperatures.

The writer believes item *b* of the author's summary in Par. 21 to be sound, only he would limit temperature to 700 rather than 750 or 800 deg. fahr.

Item *c*, the need for the new steam tables, should be emphasized.

Item *d* seems to be the key of the entire problem. If engineers were to spend more time figuring how to improve the operation of the present plant instead of trying to rebuild it, the time would be more profitably spent.

The writer is not interested in schemes for improving plant economy a few per cent and which require increased investment, as the overhead on the investment will frequently be more than the savings and the improvement may result in complications to

the plant. There are three problems to be faced; to put a kilowatt on the bus for the lowest B.t.u., to build a plant to do this at the lowest capital cost, and to obtain a plant which will run through any condition or emergency. The end sought is to serve the community at any and all times, and to do it so that stockholders get a reasonable return on their money.

Peter Junkersfeld. We are all indebted to Mr. Orrok for a classic paper on an important subject. The information that he has given us, and also that given by Mr. Morrison on properties of metals at high temperature, was much needed at this time.

The final question with any central station is always — Does it pay? Mr. Orrok has shown that for the present, temperatures of 700 to 750 deg. fahr. should not be exceeded, and in Fig. 2 has shown that the extra fixed charges are always greater than the saving of coal at $6.00 per ton for the Rankine cycle (ordinary operation) at 700 deg. maximum temperature and other conditions that he has given. It will be a comparatively simple matter to derive all but one of the curves in Fig. 2 for coal of a different price and quality. The exception is the curve of "extra fixed charges per kw-hr." The determination of fixed charges is one thing, but fixed charges per kw-hr. is quite another matter. This curve will be much higher with some companies and much lower with others, being based on three items — (1) extra construction cost, (2) percentage of interest, depreciation, etc., and (3) the kw-hr. per year over which the number of dollars per year resulting from (1) and (2) may be spread. The effect of Item (3), kw-hr. per year or annual output, is so much greater on extra fixed charges per kw-hr. that for the purpose of this discussion we may assume that Item (1), extra construction cost, and Item (2), percentage for interest, depreciation, etc., are both constant for a number of central stations whose extra fixed charges per kw-hr. we might wish to determine. We now have only one factor to deal with, namely, output, or as it has been more specifically and properly defined — generator capacity factor.

This factor, which is the ratio of the actual output to the possible output of a steam turbine as rated with all its accessories when averaged over a period of years, is much less than usually supposed, especially among central stations that have a substantial annual average increase in output.

An examination of the experience of seven companies with 14 power stations and 92 steam turbines, aggregating 1,730,000 kw. rating, shows that in substantially the same 12 months' period one had an average generator capacity factor on all its turbine units of 44 per cent and another of 25 per cent. The extra fixed charges per kw-hr. for the latter would be almost double that for the former.

It was also noted that only two of the seven companies had an annual capacity factor of 60 per cent or better on any one turbine. The company whose successive installations apparently gave improved economy and reliability and which made good use of such equipment had the following average capacity factors during a 12 months' period:

Turbines	1	yr.	in	service,	per	cent	77
"	3	"	"	"	"	"	55
"	4	"	"	"	"	"	38
"	5	"	"	"	"	"	32
"	8	"	"	"	"	"	23
"	9	"	"	"	"	"	3
"	10	"	"	"	"	"	4
"	11	"	"	"	"	"	2
				Average,	"	"	37

Only one of the companies had turbines still available for use, two in number, that had been in service 18 years, and only three companies had such turbines, 19 in number, that had been in service 14 years or more.

An average experience for these seven companies, all in large and growing cities, indicated a capacity factor of about 58 per cent on turbines one year and 30 per cent or less on turbines ten years old. The actual average of the particular five turbines that were eleven years old was only 20 per cent, but this point was below the average curve.

The falling off with age of the capacity factor of turbines reflects good operating procedure with improved equipment in the successive installations. The fact that only one company operated its one-year-old turbines at 77 per cent, another at 52 per cent, and the third at only 23 per cent, and that the average of 10 turbines two years old operated by six companies was only 50 per

cent, indicates that there is still very great room for improvement in construction and operation of equipment with what might today be called moderate pressures and superheat.

The 92 turbines considered above were all operated at pressures from 150 to 250 lb. at the throttle, and total temperatures from 475 to 600 deg. Some of the seven companies have turbines with 300 lb. pressure, but are not included because a full year's service had not been completed at the end of the year 1921, the period from which most of this experience record was taken.

In all of the foregoing wherever reference is made to turbine experience it means not only the turbine proper, but also the condensing equipment, piping and other mechanical accessories, and the electric generator, the excitation, the switching equipment and other electrical accessories.

The development toward higher steam pressures and superheat has been under way many years and will inevitably continue. It should be given every reasonable encouragement consistent with established scientific facts and sound economic possibilities. It should be remembered, however, that this development in the past in a broad and overall aspect has tended toward lower construction as well as lower operating costs of the station. The development now immediately before us seems to tend toward higher costs of construction and of operation with the exception of fuel. It thus becomes even more necessary than heretofore to determine proper balances between fuel saving and what Mr. Orrok has called "extra fixed charges per kilowatt-hour." The latter is therefore particularly important and especially because experience has shown that it varies so widely from turbine to turbine, from station to station, from company to company, or city to city, and from year to year.

A. M. Houser. With reference to the use of cast-steel material for high temperatures and pressures, it may be well, when considering values given in investigations made by Crane Co., to state that we regard these values as conservative.

All of our tests have been made by taking temperatures inside the bar being tested, and close to the section where the break occurs. The work done at the Bureau of Standards by French, the writer understands, was measured in a similar manner. Others have taken temperatures by fastening the tip of the pyrometer

couple outside and against the bar. Further, we use a " gap " coil wound on ends only, as shown in Fig 53, whereas some others use " full wound " heating elements.

We are undertaking comparative experiments to determine the difference in temperature readings taken from the inside of the bar as we have been taking them, and the outside of the bar as temperatures have been taken by some others. Some time ago,

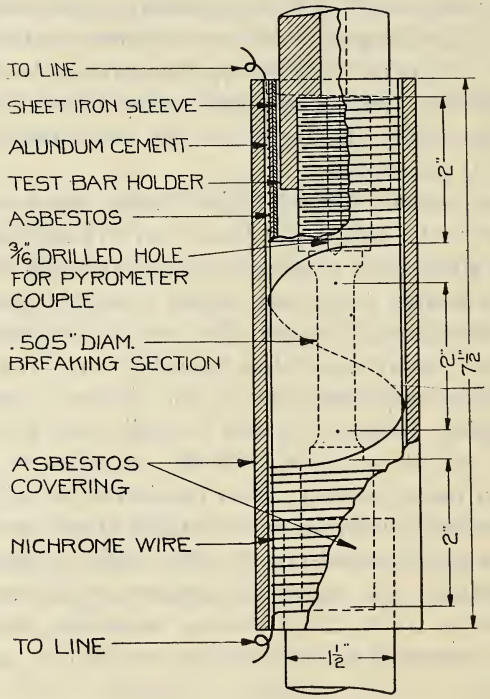


Fig. 53 Method of Heating and Measuring Temperature of Test Specimen

we made a test of this kind and found a difference in readings taken from the inside and the outside of the bar of from 30 to 100 deg. fahr., depending somewhat upon the length of time taken to heat the bar. Only one test was made and we, therefore, do not consider it conclusive; we now have other tests under way. Such differences, of course, will give from 5,000 to 15,000 lb. higher tensile strength on a falling curve such as we have on cast steel at temperatures above 550 deg. fahr.

Temperature readings taken inside the bar theoretically should be lower than those taken outside and we have always found them so. This is so because the outside thermocouple gets direct radiation heat from the heating coil and will, therefore, register high, and secondly, even without such direct radiation heat, and assuming that the thermocouple registers the outer fiber temperature of the bar perfectly, the outer fiber will have greater temperature than the center of the bar because the heat is going from the outside inwardly.

As strengths are falling in the curves of all investigators of whom we know (except MacPherran) at temperatures beyond 575 deg. fahr., any error on the high side in temperature readings will be favorable; that is, will show a higher strength than the material really has at that temperature, and errors on the low side in temperature readings taken will show a lower strength than the material really has. In other words, our own curves, if in error, err on the low side. We have preferred to have our materials safer in strength, i.e., underrated, though our curves do not look as high nor our materials as favorable as they would were the error thrown on the other side, through reading temperatures too high.

We have made further tests on cast steel. These tests, just completed, indicate the possibility of increasing these values over those given some years ago, partly through improved method of testing and partly because of improvement in quality of steel which is now made by the basic electric process.

The following gives a few results of our present investigation as compared with our former data. Referring to Fig. 54, the tensile strength curve at 600 deg. fahr., shows an increase over the old curve of about 11 per cent, and at 700 deg. fahr., about 18 per cent.

On the elastic limit curve the increase is about as follows: 600 deg., 20 per cent; 700 deg., over 9 per cent; but at 1000 deg. the increase is around 16 per cent; that is, our old test at 1000 deg. shows an average of 9650 lb., while our new test averages 25,170 lb. This figure is substantially verified by tests made at 1200 deg. which show an average elastic limit of 17,470 lb. per sq. in.

C. H. Smoot. Mr. Orrok's valuable paper brings forth very clearly that the future development of power generation by steam should lie in increased steam pressures and not in increased temperatures. The search for higher economy is plainly shown by his

TABLE 11 TENSILE TESTS OF ELECTRIC CAST STEEL, ANNEALED, CRANE CO.

Temp. deg. fahr.	70	200	300	400	500	550	600	700	800	900	1000	1100	1200
						Tensile Strength — Pounds per square inch							
Set No. 75	80500	73350	71000	70800	74450	76550	75700	68800	59800	48400	35900	26200	21700
Set No. 78	81350	73650	71325	72600	74750	76950	76000	71150	60900	50300	42450	32400	24400
Set No. 82	77200	71450	68300	68650	71500	72800	72200	67700	57800	48800	31750	29650	23100
Ave.	79680	72820	70210	70750	73570	75425	74630	69220	59500	49170	36700	29420	23070
						Yield Point — Pounds per square inch							
Set No. 75	48300	41500	41500	38000	36000	35000	35000	34500	31500	28500	23000	17750	16900
Set No. 78	42000	36500	35750	36500	35000	36500	37000	36000	32500	37000	29500	25000	18500
Set No. 82	40000	37000	35500	33000	34500	34000	34000	37500	33500	28500	23000	21000	17000
Ave.	43430	38330	37580	35830	35170	35500	35330	36000	32500	31330	25170	21250	17470
						Elongation — Per cent in two inches							
Set No. 75	27.3	28.0	28.0	27.0	22.0	24.0	19.0	34.5	35.0	35.5	29.5	28.7	27.5
Set No. 78	27.0	26.8	24.7	24.5	21.5	20.5	18.5	28.3	29.0	31.3	28.5	24.5	26.5
Set No. 82	27.0	27.5	26.0	25.5	22.5	22.5	23.3	29.0	35.5	43.5	34.0	32.0	31.5
Ave.	27.1	27.4	26.2	25.7	22.0	22.3	20.3	30.6	33.2	36.8	30.7	28.4	28.5
						Reduction of Area — Per cent							
Set No. 75	42.4	42.8	42.5	41.3	35.4	29.9	22.0	47.6	54.7	53.4	46.4	48.4	48.8
Set No. 78	43.1	41.0	38.6	43.1	34.1	24.4	19.9	43.2	50.6	58.6	52.0	42.5	45.2
Set No. 82	43.6	45.5	46.4	43.4	35.7	34.8	32.2	41.4	56.1	65.1	50.4	47.5	48.5
Ave.	43.0	43.1	42.4	42.6	35.1	29.7	24.7	44.1	53.8	59.0	49.6	46.1	47.5

thermal efficiency curves to have a fruitful field only in greater pressures.

Further increases in superheat with moderate pressures lead to moderate increases in economy and the temperature cannot be carried much higher with present materials. It seems to the writer, in view of the circumstances brought forward by Mr. Orrok, that any attempt to increase the available heat of steam by superheat, interstage re-heating, etc., is but a temporary expedient, and in

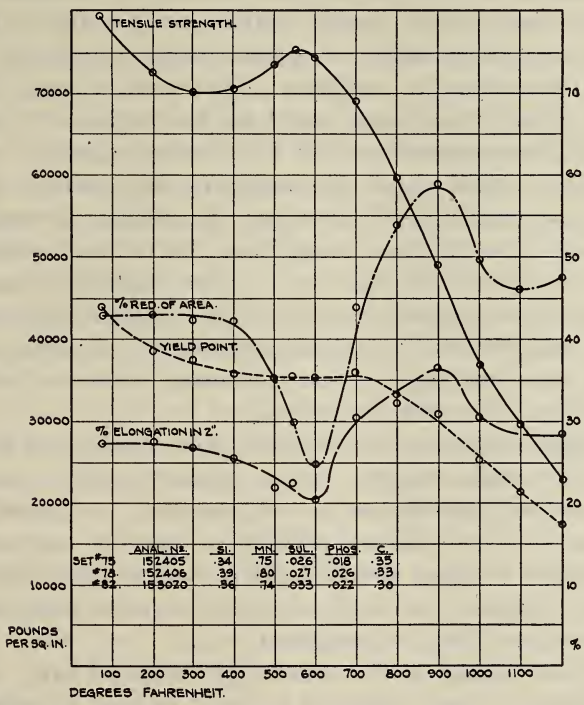


Fig. 54 Results of Crane Co.'s Present Investigations

effect is merely an evasion of the real issue and a patch whereby existing engineering on turbines and boilers may be maintained a little longer in use.

The very moderate gain to be had from further increase of superheat is totally unjustified by the very materially increased risk in the practicability of the apparatus and materials at higher temperatures. The behavior of materials of construction under increased pressure is a comparatively simple problem. On the

other hand, the behavior of materials under higher temperatures is today an uncertain matter fraught with possibilities of unanticipated events. To add temperature to the steam sufficient for a material improvement of economy calls for extended research in construction materials; further, such research is only dependable after being seasoned with use and time. To go to increased pressures without superheat calls for the addition of metal, the properties of which remain within the scope of existing knowledge.

In a general way the increase in turbine economy due to superheat must always remain moderate, since the addition of superheat invariably entails a corresponding increase in entropy and heat rejected to the condenser. On the other hand, increased pressures reduce the entropy and total heat rejected to the condenser, and the regenerative cycle still further reduces the rejected heat. While under some circumstances an increase in plant economy may result from superheat, the further development of superheat is in a direction away from the ultimate solution. A steam turbine generating plant with a given maximum temperature always will have a greater economy when operating with saturated steam at high pressure than when operating with superheated steam and also with interstage re-heated steam, where the re-heat is not taken from otherwise waste heat.

It is high time that the problem of the regenerative-cycle turbine be thoroughly studied. In the writer's opinion the present type of turbine construction is most decidedly not suited to the application of this cycle and possibly is not the best suited for central station work on the heat cycles now in use. The lack of suitability, however, is much more pronounced than when the regenerative heat cycle is considered.

A steam turbine of the present conventional type, in which the admission of steam sufficient to carry its load is controlled at the high-pressure end, has throughout its body pressures varying roughly in proportion to the load. This makes the effective application of a regenerative cycle difficult, for the places in the turbine from which heat must be taken and added to the counter current flow of condensate do not have a fixed pressure and temperature, but one which is continually changing. As the load becomes less, the amount of heat added to the condensate and its ultimate temperature would become less and in a somewhat greater ratio than in proportion to load. As a result, the most effective application of the regenerative cycle to the present type of turbine con-

struction involves a condition of nearly uniform load on the turbine, which is hard to realize in plant operation.

A second difficulty resides in the water content of the steam, which becomes excessive in the lower pressure stages when steam is expanded from an initial pressure higher than in current use.

There is no solution for this problem without a change in turbine design, inasmuch as the higher economy is dependent upon the expansion producing a higher percentage of moisture. The efficiency of a steam turbine is adversely affected by moisture. The action is twofold. First, the particles of water retard the flow of steam, resulting in a flow composed of heterogeneous average of dry steam and entrained water particles. Losses result from the expansion of the steam acting by shock to accelerate the particles of water. Second, a portion of the water is separated where the steam passes through the nozzles and again where it makes the curved flow through bucket channels. Water collects on the concave surfaces and is rubbed off by the high velocity steam, again producing shock loss.

To realize maximum economy in the high-pressure turbine, it is very evident that new means must be found of abstracting the water which is formed by expansion. It is obviously a serious mistake to pass through the low-pressure section of a turbine steam containing some 25 per cent moisture, and turbine designers must find means of abstracting the water between stages. This condition again calls for a change in turbine construction.

A turbine of large power, as employed in central-station practice, should have a wide, high, efficient range of operation; and an improved economy on light loads is a very necessary condition. With the lower load factors incidental to public service station operation, economy at light loads throughout the plant is important.

Even aside from the design of a turbine for high steam pressures, the type of turbine now in use is markedly deficient in its behavior at light loads, but with the regenerative cycle and present type of turbine construction, the light-load economy would be proportionately less than is at present the case at moderate pressures, for the reason that the heat recovery by counter-current condensate heating would be reduced on light loads, so that to the natural loss in economy of the turbine alone would be added a further loss in heat cycle efficiency from a reduction in the heat recovery of the regenerative cycle on light loads.

The matters to be considered in adapting a turbine to a high-pressure regenerative heat cycle have only been partially enumerated, but the writer believes the above are the more important.

There is one other element which must not be overlooked, and it is our lack of knowledge of the steam tables in the higher pressure range.

The extension of our knowledge in this region might indicate an even greater improvement in thermal economy by the use of high-pressure saturated steam than is now deduced from our admittedly inaccurate data.

While there are serious problems to be met in the application of a regenerative heat cycle to the present conventional type of turbine, the problem should not be unsolvable, and may be answered by a change in design.

In considering the regenerative cycle as applied to a turbine in a power plant, a question arises at what point to stop the regenerative heating of the condensate in order that the condensate may be at a temperature when leaving the turbine sufficiently low to be still capable of taking up a reasonable amount of heat from the economizer or its equivalent extension of the boiler. It is, of course, no use to employ regenerative heating in the turbine and waste sensible heat in the flue gas, and the extent to which the regenerative heating is carried through by the turbine must be limited in order that the flue-gas heat be used.

These conditions are not in conflict and the most useful part of the regenerative cycle is at the lower temperature range, this gain in efficiency due to the regenerative cycle becoming small at the high-pressure end of the turbine. On the other hand, the economizer heating of the condensate between turbine and boiler is all gain in thermal economy.

In practice, the writer believes that the regenerative feed-water heating will be limited to a maximum temperature of from 250 to 300 deg. fahr. and then heated up to steam temperature in the economizer section of the boiler. This would seem to be perfectly practicable and offers a still further improvement in plant thermal efficiency than is obtainable from a regenerative cycle without the use of the economizer. The addition of regenerative cycle and economizer should still make it possible to bring the flue-gas temperature below 500 deg. fahr. and at the same time realize the advantage of the regenerative cycle.

W. J. Wohlenberg. In this valuable contribution the author arrives at expected gains resulting from a large increase, principally in pressures, when operation is based on the Rankine cycle and on the regenerative cycle having a number of bleeding stages.

The heat-balance cycle commonly used in modern power plants is, of course, of the regenerative type even though the high-pressure steam is divided between auxiliaries and main units, in which, however, only one bleeding stage is employed. Based on the total power developed this cycle will have efficiencies lying between those of the Rankine and multi-stage regenerative cycles. Performance data from actual practice for the heat balance plant are represented in Table 1 of the author's paper. It will be instructive to have figures for it on the same basis as that used for computed data shown in Tables 2 and 3.

It is to be noted also that the computed results are based on an expected realization of the same high boiler and turbine efficiencies throughout but the means of accomplishing this is not indicated. As a matter of fact a decided increase in boiler pressure will, due to higher surface temperatures, very materially increase the escaping gas loss unless economizers or some other devices for reclaiming heat from the flue gases are installed. Economizers may not be used to full advantage in multi-stage regenerative stations as the feedwater has its temperatures increased by heat supplied in steam bled from the main turbines. At 200 lb., boiler efficiencies of 80 per cent are obtainable at reasonable ratings by means of high or deep boilers, but at 1200 lb. the boiler surfaces have increased in temperatures to nearly 600 deg., resulting in relatively high escaping gas losses. For such conditions in stoker-fired plants it will be difficult to realize boiler efficiencies even up to 75 per cent. In multi-stage regenerative plants flue air preheaters would serve to increase the steam generator efficiency. Air preheaters require very large surface areas because only low rates of heat transmission are possible without excessive draft losses and are, furthermore, at the present time, of undecided merit, at least as far as American practice is concerned.

Regarding the turbine, it is doubtful if with pressures of 1200 lb. it will be possible to realize referred efficiencies equal to those existing or to be obtained at lower pressures unless very high superheats can be maintained. Even at an initial steam

temperature of 1000 deg. with the above pressure it appears that the dew point occurs relatively no later in the expansion than is the case for present-day operation, and with 750 deg. it will occur earlier. Under such conditions both blade losses and maintenance in the low-pressure sections will probably increase. The construction of the high-pressure stages appears to leave only a choice between relatively inefficient arrangements. If reaction blading is used the leakage and rotational losses increase and if these losses are maintained low by great pressure reductions in first-stage nozzles it is necessary to use the relatively inefficient velocity compounding wheel. It seems, therefore, that it is not quite fair to units operating on the lower pressure range if equal referred turbine efficiencies are assigned to proposed units operating through a much greater pressure range.

Taking these conditions into consideration new values are computed and exhibited in Table 12 of this discussion. Statements are included concerning the extent and arrangement of heating surface necessary for attaining the given steam generator efficiencies and computed performance figures on an equal basis are shown for the one-stage regenerative or heat-balance cycle as employed in modern power plants.

From this table comparisons of various arrangements are arrived at in which the expected fuel saving in each case is charged with the extra equipment required. The results of this comparison are shown in Table 13 of this discussion.

The new table of results brings out three separate conditions each of which tends to decrease the margin of gain shown for the regenerative cycle at high pressures in Fig. 2 of the author's paper. First, the heat-balance cycle has a considerably higher efficiency than the Rankine cycle used as a basis for computing the gains; secondly, the efficiency of the regenerative cycle appears slightly lower because of the reduced turbine efficiency, and finally either the overall efficiency will be reduced because of a lower boiler efficiency or the additional requirements in heating surface will increase the cost and extra fixed charges over the values shown. It is at least not apparent in the paper that the latter have been taken into account.

The net result of the revised figures on the commercial aspect is well shown by comparing items 4 and 8 in Table 13 of this discussion. The author's figures show a saving in fuel of 30.8

TABLE 12 COMPARISON OF PROPOSED CYCLES WITH 200 AND 1200 LB. PRESSURE

Case	Pressure lb. per sq. in.	Temp. deg. fahr	Cycle	Steam generator*	Comb. eff. steam gen.	Ref. eff. steam turbine	Eff. elec. gen.	Radiation piping and auxiliaries	Station eff.	Theoretical eff. steam cycle	Overall ther. eff.	B.t.u. per kw-hr.
1	200	750	Rankine	Boiler	0.80	0.78	0.97	0.91	0.55	0.335	0.184	18,500
2	1200	750	Rankine	Boiler and 50 % econ.°	0.80	0.75	0.97	0.91	0.53	0.405	0.215	15,880
2A	200	750	Heat balance	Boiler	0.80	0.78	0.97	0.91	0.55	0.357	0.196	17,400
2B	1200	750	Heat balance	Boiler and 50 % econ.°	0.80	0.75	0.97	0.91	0.53	0.438	0.232	14,700
3	200	750	Regenerative	Boiler	0.80	0 78	0.97	0.91	0.55	0.370	0.204	16,800
4	1200	750	Regenerative	Boiler	0.75	0.75	0.97	0.92	0.50	0.485	0.243	14,080
4A	1200	750	Regenerative	Boiler and 100 % flue° air preheater	0.80	0.75	0.97	0.91	0.53	0.485	0.257	13,280

* Boiler surface absorbing heat at an average rate of 6000 B.t.u. per sq. ft. per hr. not including economizer. Fuel burned on stokers.

° Percentage based on boiler surfaces.

TABLE 13 COMPARISON ON FUEL CONSUMPTION AND EQUIPMENT BASIS

No.	Steam temperature throughout 750 degrees.		
	Conditions compared	Computed reduction in fuel consumption	To be charged against reduction
1	Heat-balance plant with economizers at 1200 lb. to Heat-balance plant at 200 lb.	15.5 per cent	Extra cost of high-pressure equipment plus 50 per cent additional surface in economizers
2	Regenerative plant at 1200 lb. to Regenerative plant at 200 lb.	16.2 per cent	Extra cost of high-pressure equipment
3	Regenerative plant with air preheater at 1200 lb. to Regenerative plant at 200 lb.	20.9 per cent	Extra cost of high-pressure equipment plus 100 per cent additional surface in flue air preheaters
4	Regenerative plant at 1200 lb. to Heat-balance plant at 200 lb.	19.1 per cent	Extra cost of high-pressure equipment
5	Regenerative plant with air preheater at 1200 lb. to Heat-balance plant at 200 lb.	23.6 per cent	Extra cost of high-pressure equipment plus 100 per cent additional surface in air preheaters
6	Regenerative plant at 1200 lb. to Heat-balance plant with economizers at 1200 lb.	4.2 per cent	A negative charge as heat-balance plant contains 50 per cent more surface in economizers
7	Regenerative plant with air preheater at 1200 lb. to Heat-balance plant with economizer at 1200 lb.	9.6 per cent	Difference in cost of 100 per cent air preheater and 50 per cent economizer
8	Regenerative cycle at 1200 lb. as computed by author to Rankine cycle at 200 lb.	30.8 per cent	Extra cost of high-pressure equipment plus at least 150 per cent surface in air preheater

per cent by an increase of pressure if the gain is based on a Rankine-cycle performance at the lower pressure, whereas, based on the heat-balanced cycle performance at the lower pressure the revised figures show a saving for the high-pressure regenerative cycle of only 19.1 per cent. This will reduce the coal saving from

0.257 to 0.154 lb. per kw-hr., Table 3 of the author's paper, and the money saving by fuel from over 10 cents to less than 5 cents per kw-hr. Thus even for this case the extra fixed charges appear actually to outweigh the saving in fuel.

Before coming to definite conclusions let us not, however, lose sight of the fact that as more high-pressure equipment is developed its relative first cost, and hence the relative fixed charges, will probably decrease, that in the long run the relative price of fuel will probably increase, that between two ways of making power requiring equal effort and equal total expenditure that one is best which conserves the most fuel, and finally that high-pressure steam properly applied appears even in the present undeveloped state of the high-pressure art to very nearly hold its own. With these facts in view the following conclusions are stated:

1. The extra cost of fixed charges in high-pressure equipment will just about offset the saving in fuel consumption due to the use of higher pressure steam in stoker-fired plants. These extra fixed charges would be reduced in amount for powdered-fuel plants due to the higher possible rate of driving the boilers for a given efficiency.

2. The possible saving in fuel consumption is considerable and should encourage manufacturers to attempt the development of high-pressure equipment at lower costs.

3. The highest efficiencies may be obtained in high-pressure plants operating on the regenerative cycle and having air preheaters installed to reclaim heat from the flue gases after leaving the boiler surfaces.

4. The regenerative cycle appears so advantageous at any pressure that its use should always be seriously considered.

5. With high pressures either economizers or flue air preheaters will be necessary if high efficiencies are to be realized in the steam generators. Hence high pressures will force the development of this equipment.

6. If air preheaters are found impracticable, thus forcing lower steam generator efficiencies in high-pressure plants operating on the regenerative principle, it appears that it will still be advantageous to use some form of the regenerative cycle in which the feedwater at a fairly high temperature enters the economizer. In short, there is probably some best point which involves a certain temperature of feedwater below that of the boiler making possible

lower flue-gas temperatures by use of a limited amount of economizer surface.

D. S. Jacobus. The paper is a most valuable one and the author should be congratulated for the broad way in which he has handled the problem.

The additional efficiency secured through operating at higher steam pressures and higher temperatures of steam is well brought out and the conclusions in the summary show the possibilities with the materials ordinarily available. Speaking from the boilermaker's standpoint, Mr. Orrok is correct in saying that pressures up to 1200 to 1500 lb. may be attained, whereas there are special features of design that must be carefully considered when it comes to the construction of a unit of large size for power plant work, and no large sized units for these pressures have so far been constructed. This would be no serious problem as the Babcock & Wilcox Company is now constructing such boilers for 650 lb. working pressure. He is further correct that temperatures up to 700 to 750 deg. should not be exceeded. It is true that steam can be readily superheated to higher temperatures than this and is employed in some cases, whereas with our present knowledge of valves and fittings it would seem that 750 deg. is a good figure to set as a maximum for large power plant work.

The building of a boiler and steam turbine for the higher pressures and superheats is, as brought out by the author, not all of the problem. The type of valves and pipe fittings to employ must be considered. The increase in steam pressures has proceeded so rapidly that there is no standard for flanged fittings for the higher pressures. A number are working actively on designs which eventually will come before the proper committees of the Society for adoption as standards. Flanges for steam fittings have not been standardized for pressures in excess of 250 lb. per sq. in. A statement is made in the Report of the Committee of this Society on Standardization of Flanges and Pipe Fittings [1] that flanges constructed in accordance with a table it prepared for 800 lb. hydraulic pressure should be all right for 400 lb. steam pressure and steam temperatures not exceeding 250 deg. superheat. The arrangement for the hydraulic flanges is not, however, what it should be for high-pressure steam flanges on account of the great amount of overhang of the bolts.

[1] Trans., vol. 40, p. 501

The results of tests on the properties of metals at high temperatures given by Mr. Morrison show clearly that there need be no apprehension respecting the ability of the piping to withstand the pressure. The falling off in the elastic limit at high temperatures is, however, much greater than the falling off of the ultimate strength and this affects the design of flanges, as with too great an overhang of the bolts the flanges may become distorted, due to the initial strain and due to the strains caused by the alternate expansion and contraction. Another feature that should be borne in mind is that if the bolts are placed too far from the pipe, so that they may be at a considerably lower temperature than the pipe, it is entirely possible to strain the bolts beyond their elastic limit through the difference in expansion of the flanges and the bolts. All features point to the advisability of making the bolt circle as small as it can be made and be consistent with good design.

Another feature that should be borne in mind in designing flanges is that the bolts may be set up so hard that something will have to give. It is necessary, therefore, to decide what should be the weakest element in case of over-strain. Off-hand, it would seem that it would be best to make the bolts the weaker element, as in case of over-strain, they could be readily replaced, whereas should the bolts be made strong enough to dish and distort the flanges, the flanges would have to be replaced. Of course, the bolts should be very much stronger than required to hold the flanges together and to make a tight joint under ordinary conditions.

There is a need of data to show whether the elastic limit of steel at high temperatures will be permanently elevated in the case of an over-strain in the same way as at ordinary temperatures. Experiments on this feature would be most useful.

Mr. Jacob Perkins was a pioneer in the use of steam at high pressures and high temperatures. Mr. Perkins' work is described in the History and Progress of the Steam Engine by Elijah Galloway, published in London in 1830. Mr. Perkins advocated heating the water to a high temperature under a sufficient pressure to prevent boiling, and then passing it through a throttle valve and flashing it into steam at a lower pressure for use in a steam engine.

A steam generating apparatus patented by Mr. Perkins in 1824 is described in Mr. Galloway's book. This was made up of cast-iron bars 5 in. square with circular holes $1\frac{1}{2}$ in. diameter. The hollow bars were arranged in three horizontal rows over the fire.

Water was forced into the two upper rows, entering the hollow bar most remote from the fire and passing successively through each bar. After being heated in the two uppermost rows the water passed through a throttling device into the lowermost rows of hollow bars, which were subject to the direct radiant heat of the fire and kept at a temperature of about 1000 deg. fahr. At each stroke of the engine a certain quantity of water or steam and water heated to 700 or 800 deg. fahr. was discharged into the lowermost row where it was flashed into steam and superheated. From the lowermost row of hollow bars the superheated steam passed to a steam chamber in which the steam could be maintained at any desired pressure, and from thence to a steam engine. The pressure in the water-heating elements of the Perkins boiler must have been that corresponding to the critical temperature or over to allow the water to be discharged at from 700 to 800 deg. fahr., so that it must have been at least 3000 lb. per sq. in.

V. T. Malcolm [1] and Charles R. Phillips.[2] We agree with Mr. Orrok that engineers are now using better and more uniform materials than ever before, and that the manufacturer has kept pace with these conditions and is endeavoring to assist the engineer in the metallurgical and manufacturing processes.

In this connection we mention the manufacture of steel castings in our plant by the acid electric-furnace process, not because the electric-furnace process is cheaper in cost than the open-hearth or Bessemer process, as it is a well-known fact that electric-furnace steel is rarely lower in cost than open hearth and never lower than Bessemer, therefore the reason for its use is due to its producing a superior steel.

Dr. John A. Mathews,[3] President of the Crucible Steel Company of America, stated in a paper read before the American Iron and Steel Institute at its New York meeting, October 27, 1922, the result of actual count of hairlines due to inclusions in the ground surface of steel of the same chemical specifications. As the result of tests on several heats of steel by the basic and acid open-hearth process and the electric-furnace process, the average count ran in a ratio of 8 to 4 to 1, and the hairlines in electric-furnace steel were much shorter than in the steel made by the open-hearth process.

[1] Metallurgical Engineer, Chapman Valve Mfg. Co., Indian Orchard, Mass.

[2] Sup't Foundries, Chapman Valve Mfg. Co., Indian Orchard, Mass.

[3] *Chemical & Metallurgical Engineering,* vol. 27, no. 18.

The better quality of acid electric-furnace steel is largely due to the maintenance of a reducing or nonoxidizing atmosphere within the furnace, resulting in a more nearly pure and completely degasified steel.[1]

Fig. 55 shows the average analysis of electric-furnace steel and the method of keeping the records of heats.

As the production of sound steel is the first requisite, the second, and of the same importance, is the proper and thorough

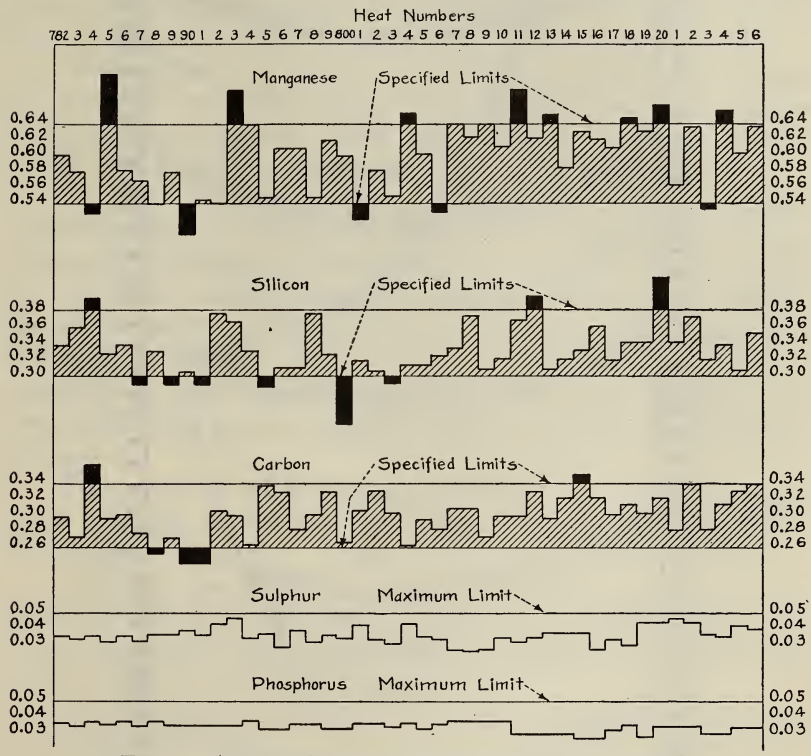


Fig. 55 Average Analysis of Electric-Furnace Steel

annealing of the castings, so that the castings may possess the best physical properties of which steel is capable. High-grade steel castings may be poorer in quality through improper annealing than an inferior steel which has been properly annealed. Knowledge of the critical points and transformations is required in order to secure the highest physical properties.

[1] Moore, Trans., vol. 41, p. 971.

Ordinarily the finer the grain the stronger the steel, and to obtain this refinement the heating and cooling cycle for steel must be determined as narrowly as possible.

It naturally follows that the operator must have the facilities which eliminate, so far as possible, variables in carrying out the

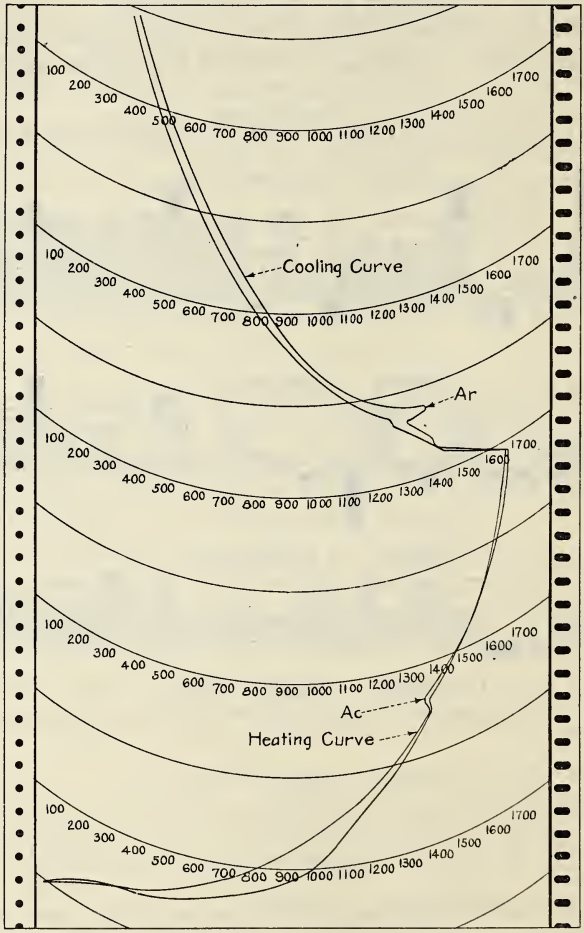


Fig. 56 Transformation Points 0.26 Carbon Electric Steel

process required, if uniformity of product and full worth of steel is to be secured.

Each heat of steel is carefully analyzed as to its chemical composition, and in addition a thermal analysis made, so that the transformation point may be determined, as illustrated in Fig. 56.

The steel is treated by heating to well above its critical points, and soaking for proper length of time so that thorough penetration of heat may be obtained and then cooling at proper rate to a temperature below redness; thus breaking up the networks and large crystals into fine-grained structures.

The purpose of annealing the castings is to increase the softness and ductility, and to eliminate the existing coarseness of grain, and to secure the proper combination of ductility and elasticity for resisting strains, and to relieve internal strains such as may be caused by uneven contraction of castings in the mold.

Steel castings are annealed at this plant in an electric furnace of the nichrome resister type, which is controlled by a Leeds-Northrup potentiometer, which plots a chart of the full history of the heat, and these charts are always kept for record.

The electric furnace of the resistance type affords an atmosphere free from contaminating gases, and the penetration of heat is even and thorough without burning or cracking, which cannot be well duplicated in a fuel-fired furnace without highly heating the outside surface and oxidizing the skin.[1] The temperature distribution is uniform, and the rate of heating and cooling is controlled precisely by the automatic control. The heating cycle once established may be repeated, insuring absolute uniformity of product.

The automatic control eliminates the human element, because when adjusted to the desired temperature it will maintain it indefinitely.

Test bars for both tensile and cold-bend tests are made in duplicate for each heat of steel, and are annealed with their respective castings, after which one set is tested in the laboratory and the other set aside for reference.

Fig. 57 shows a chart of the physical properties of steel after annealing.

With reference to the statement that modern steels allow pressures up to 400 lb. with a maximum temperature of 750 deg., and that in certain distillation plants temperatures as high as 1100 deg. have been obtained, but life of material is short, our experience and tests have shown that a properly designed valve, made from the proper materials and with a modern process, will withstand a temperature of 900 degrees without a great loss in strength and

[1] E. F. Collins, *Proceedings* A.F.A. 1920.

that deformation of such a highly unsatisfactory character as to render them useless, does not take place at 750 deg. as stated.

The valve made by our company has two axes of symmetry and is spherical in design, having a male and female joint between body and bonnet to eliminate gasket troubles.

Commenting on the physical properties of steels used in power-plant construction, Mr. Orrok states that cast steels of proper carbon content and suitable for fittings and valves show an increase in tensile strength up to 600 to 700 deg. fahr., and lose much in ductility, but above 800 deg. the tensile strength rapidly falls off and ductility largely increases. Mr. Orrok states that the lack of ductility or increase in brittleness at maximum temperatures must be compensated for by a larger factor of safety (i.e. increased

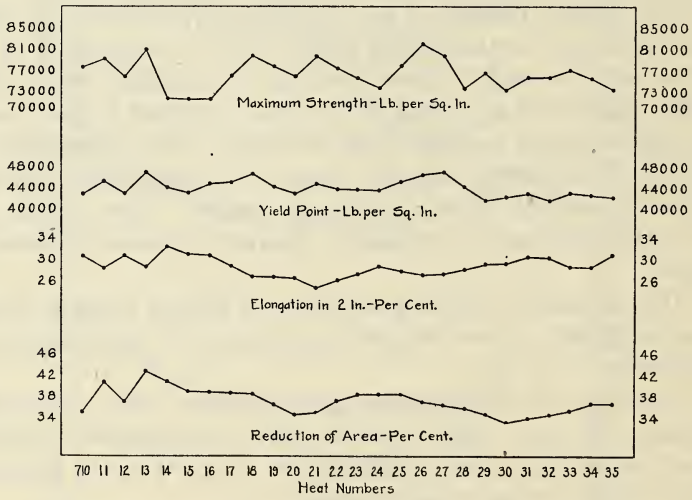


Fig. 57 Physical Properties of Steel after Annealing

thickness and weight) with higher cost and more careful selection of materials. We do not agree with Mr. Orrok with regard to this statement, for the reason that we have carried on considerable investigation into the effect of elevated temperatures on the physical properties of cast electric-furnace steel, because we felt that this was an important item for the designing engineer, on account of there being such little information available on cast steel.

Therefore, research was begun to ascertain the physical properties of electric-furnace cast steel at temperatures from 70 to 1000 deg. Considerable investigation was made of previous meth-

ods that had been used in determining this information, with the result that furnace of type shown in Fig. 58 was designed.

A is an alundum tube upon which is wound the heating element *B*. This heating element is connected to the secondary of a transformer. The tube *A* is held in the container *C* and the space between *D* is filled with kieselguhr. The specimen under test *E* screws into the specimen holders *F*, which in turn screw into the testing machine adapters *G* and *H*. *G* is the fixed member and *H* the moving member. When the specimen is in position, the doors *I* are closed and the thermocouple *J* placed against test specimen

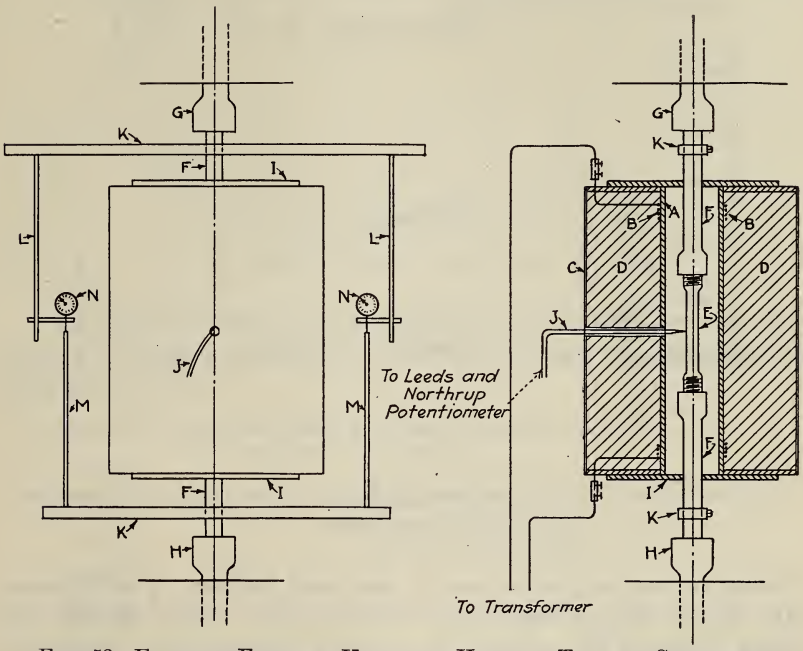


Fig. 58 Electric Furnace Used for Heating Tensile Specimens

at its mid-point. The doors *I* are made of asbestos board and fit snugly around the holders *F* so that no air currents can get into the heating chamber. As a further precaution against air currents, asbestos rope is wound around the holders *F* just inside the door *I*. The thermocouple is made of 0.025 in. diameter chromel-alumel wire, and is used with a Leeds-Northrup potentiometer. Arms *K* are rigidly fastened to specimen holders *F*. The rods *L* are fastened to the upper arm *K* and rods *M* to the lower arm *K*. Ames gages are connected to the fixed rods *L* and rest on moving rods *M*.

The test specimen is inserted in the heating unit and testing fixtures described. After the specimen has been at heat at least one-half hour, an initial load of 100 lb. is applied in order to eliminate all lost motion. The Ames gages are then set at zero, after which equal increments of load are applied and simultaneous reading of gages recorded.

The increment is 500 lb. and observations of the elongation per increment are recorded until the yield point is passed. The maximum strength, the per cent of elongation and per cent of re-

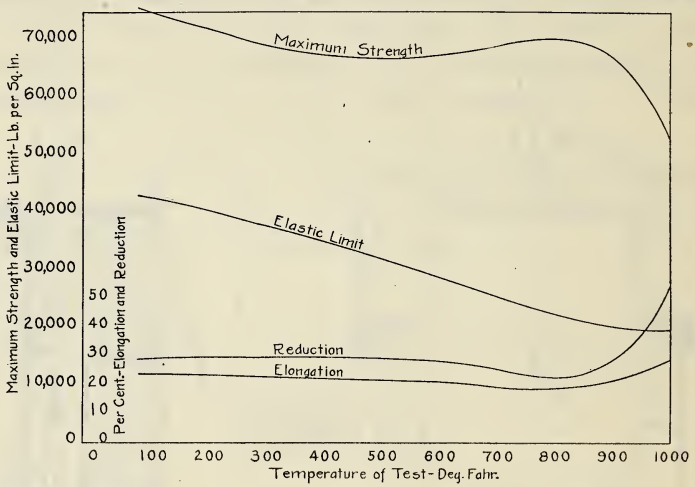


Fig. 59 Effect of Elevated Temperatures on the Tensile Properties of Cast Steel

duction of area are determined in the usual manner. The Johnsonian [1] elastic limit is determined from the curve drawn through the points obtained by plotting the average elongation per increment against the respective load.

The apparatus shown in Fig. 58 is used in duplicate, so that while one specimen is being pulled another is heating.

Fig. 59 shows the physical characteristics of electric steel at elevated temperature, while Figs. 60 and 61 show the stress-strain curves of bars used in test.

The analysis of this steel was:

[1] Materials of Construction, Johnson.

Carbon	0.254
Manganese	0.67
Silicon	0.31
Phosphorus	0.038
Sulphur	0.047

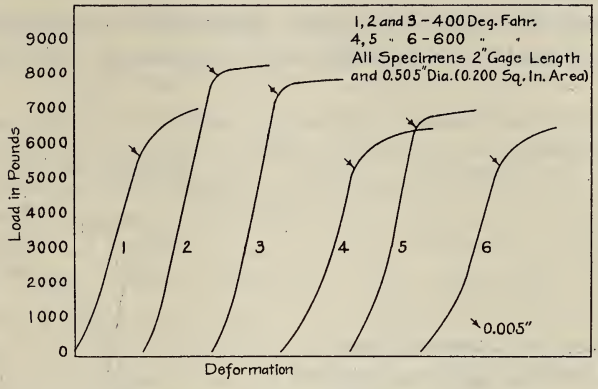


Fig. 60 Electric Steel Stress-Strain Diagram

In the microscopic examination of steel at 1000 deg. temperature we found no evidence of structural change due to testing at high temperature.

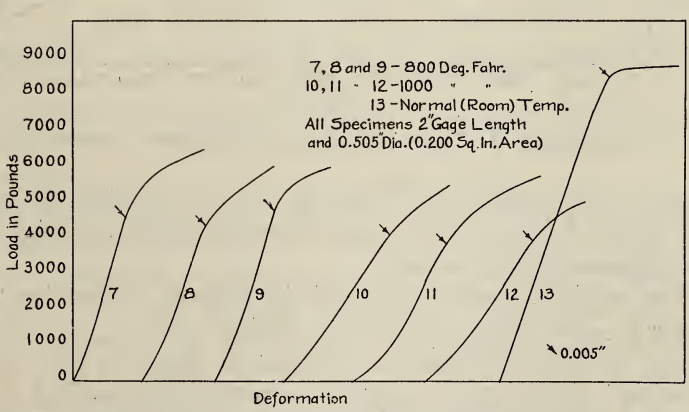


Fig. 61 Electric Steel Stress-Strain Diagram

The various investigations of the tensile properties of materials at elevated temperatures have been carried out with the use of equipment of much the same design as illustrated in this dis-

cussion. The methods of introducing the thermocouples differ in most of these investigations, however, but we believe that the area represented by the outer layers is much greater than represented by the center, and the location of the thermocouple in our investigation gives a better indication of the average temperature of test specimen.

During the tests described, temperature explorations were made, and while it is possible that a correction factor might be

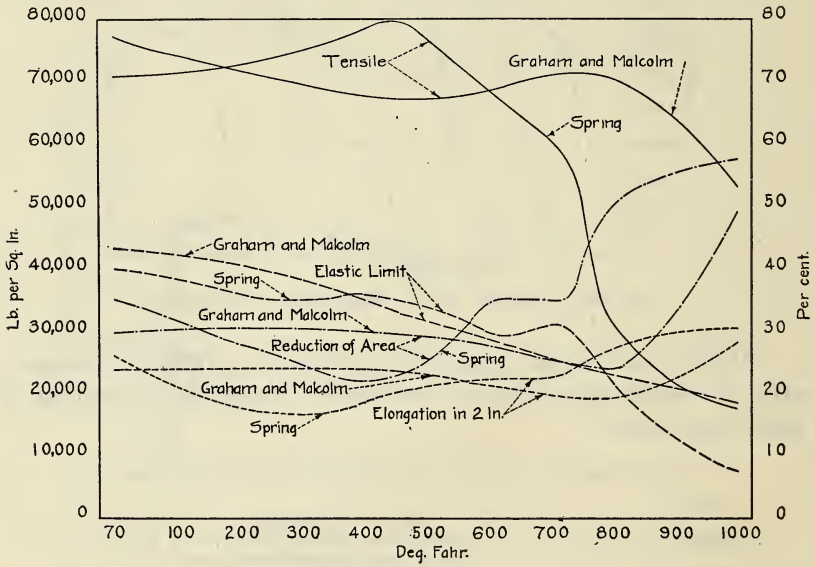


Fig. 62 Tensile Tests on Cast Steel at Elevated Temperatures

applied to the temperature readings, it is a question as to the advisability of its use as it might be considerably outweighed by other experimental variables.

The sensitive couples used and the method of applying them, we believe, gives the most accurate results.

The previous well-known information given on this subject by Spring[1] showed that maximum strength was reached at 450 deg., after which there is a sharp decline to 18,000 lb. per sq. in. at 1000 deg. Fig. 62 shows the comparison between Spring's data and our tests.

[1] Investigation into Effect of High Temperature on Metals, Crane Co. 1919.

In the recent paper by MacPherran[1] of the Allis-Chalmers Company, the maximum strength was found to be at 700 deg. which our results confirm.

The change in elastic limit is by far the most important of all changes produced by elevated temperatures, so far as valve steels are concerned, as it is shown from our diagram that the "mean variation in the elastic limit" curve continuously descends from a 70 deg. temperature, the results falling in a curve which is nearly a straight line, while the ultimate strength curve has a maximum point at 750 deg. after which it regularly decreases also.

In general the result of our test agrees very well with recent investigators, such as Professor Hoyt[2] and Mr. MacPherran.

It will therefore be seen from the results given in Fig. 59 that the ductility of our steel is not greatly affected by brittleness at 800 deg. or the maximum strength at high temperatures as referred to by Mr. Orrok, as we show an elongation of 19 per cent and reduction of area of 23.5 per cent which closely corresponds with the A.S.T.M.[3] specifications for medium steel at ordinary temperature the minimum elongation being 18 per cent and reduction of area 25 per cent.

One neglected phase of the study of the corrosion of metals has been the corrosive action of gases upon metals exposed to temperatures higher than the ordinary, among which are steel valves designed to carry superheated steam at great pressure and high temperature. Very few metals may be classed as nonoxidizing and the base metals of which steel valves are made are the oxidizing metals, and it is therefore necessary in order to attain the minimum oxidation at high temperatures to have a dense metal to resist the oxidizing effect of these high temperatures.

A search of available literature on this subject has been meager and little or no data on this subject has been published. Under certain conditions of high temperature a compound different from iron rust is formed and this compound is known as scale. The chemical composition of this scale is very complex, but it is known that it is a compound of iron and oxygen. In our experimental work we endeavored to determine the temperature at which scale begins to form in electric-furnace steel used in the manufacture of

[1] *Proceedings,* A.S.T.M., 1921.

[2] *Metallography,* vol. 1.

[3] A.S.T.M. Standards, 1921.

our valves when subjected to temperatures in excess of 800 deg. fahr.

In the experimental work bars 1 in. square were carefully ground on an emery wheel so as to remove any foreign substance adhering to them. These bars were then carefully calipered and weighed, after which they were subjected to one hundred hours' treatment in an electric furnace in which was maintained ordinary atmospheric conditions. In order to maintain this uniform rate of oxidation, the bars were removed approximately every six hours, and the specimens scraped free from scale and weighed after each cooling. Fig. 63 shows a graphical chart of the scaling weight in

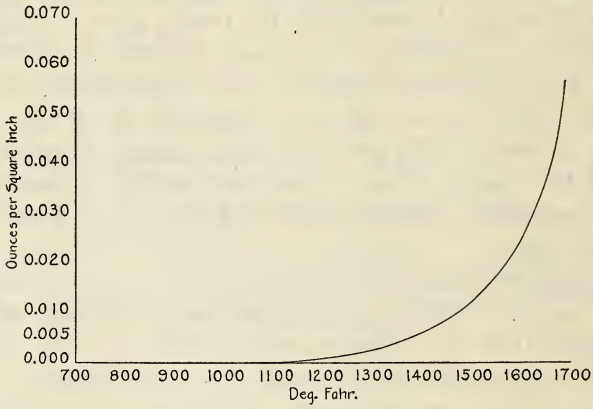


Fig. 63 Scaling Rate of Electric Steel

ounces per square inch at the various temperatures from 1050 to 1700 deg. fahr.

It will be noted that little or no scaling existed when exposed to temperatures below 1200 deg. Secondly, the formation of scale started at a temperature just above 1200 deg. and from that point increased rapidly, while at 1700 deg. the amount of scale formed may be considered excessive.

The analysis of the steel under test was:

Carbon	0.29
Manganese	0.55
Silicon	0.31
Phosphorus	0.036
Sulphur	0.033

This steel was melted in an electric furnace, and was cast into dry-sand molds, after which it was annealed in an electric furnace at a temperature at 1630 deg. for 2½ hr. and quenched in air.

The investigation was carried out by the use of test bars 6 in. long and 1 in. square, these bars being subjected to temperatures of 1000 and 1700 deg. fahr. respectively in an electric furnace under ordinary atmospheric conditions. The length of the heating period was 350 hr. after which bars were removed and allowed to cool.

These bars before being placed in the furnace were carefully measured as to length, width and depth, and after treatment for the specified period at the respective temperatures they were again carefully measured. We found an elongation in the 6 in. of length to be 0.012, there being no change in the lateral dimensions, at 1700 deg. temperature. At 1000 deg. fahr. we found no change in either length or other dimensions. The analysis and heat treatment of these bars were identical with those used as shown in scaling tests.

It is therefore apparent to us that no growth exists in the steel castings up to and including 1000 deg.

As far as we know there has been only one series of tests carried out on steel at a high temperature[1] carrying a constant load, and we decided to check these results for our own use.

In our experimental work the test specimens which were the standard 0.505 bar 2-in. gage length were placed in an electric furnace especially designed, which was attached to a 50,000 lb. Riehle testing machine and load applied. The temperature was then raised at a uniform rate until a maximum at 1100 deg. fahr. was reached and a load of 21,000 lb. per sq. in. placed on the bars. This temperature and load were maintained for a period of 400 hr., after which the bars were removed and the original gage lengths remeasured. We found no change in the original gage length or in the diameter of the bar, showing that no deformation had taken place at a temperature of 1100 deg. fahr. with a constant load of 21,000 lb. per sq. in. suspended on these bars.

The steel used in this test was the same as used in the tests as shown in scaling tests.

The bolt steel used in this investigation is made from an alloy steel known as Hy-Ten steel, and is the result of twenty years of

[1] J. H. Dickenson, Sheffield, Eng., *Engineering,* Sept. 15, 1922.

metallurgical research and practical development, with the object in view of obtaining a steel which combines the strength and toughness of high-grade chrome, nickel and vanadium steels, together with free machining qualities sadly lacking in these steels.

It has been a difficult problem to solve, but by constant study of details, by gradual development and elimination, an ideal steel for this work has been obtained. This steel itself is a chrome-manganese alloy possessing remarkable strength and toughness, and of the following analysis:

Carbon	0.44
Manganese	1.25
Silicon	0.19
Sulphur	0.02
Phosphorus	0.025
Chromium	0.52

It is therefore evident that we have in this material a steel which has solved the difficult problems of bolting for use in connection with superheated steam at high pressures and temperatures, and we know that the steel is constantly uniform in quality as the result of our numerous tests.

The grade of steel used in these bolts is known as 0.50 carbon alloy steel, No. 3 analysis and B temper, and possesses the maximum degree of strength, toughness and high resistance to shock and repeated stresses.

In heat treatment this steel is heated from 1450 to 1475 deg. fahr. and quenched in oil. The question of the drawing temperature depends upon a number of variables, and this temperature must be determined for various parts. We have found that drawing at temperature of 1200 deg. fahr. was most suitable to obtain the physical properties required, which are:

Maximum strength, lb. per sq. in.	145,000
Elastic limit, lb. per sq. in.	110,000
Elongation in 2 in., per cent	20
Reduction of area, per cent	49

In torsion, the apparent outside fiber stress at maximum load was 98,000 pounds.

As a special test[1] a rod $1\frac{5}{8}$ in. round, 14 in. long, was made

[1] Published data.

from steel of the above specifications. This rod was placed in a press with a block under each end and a force of 30 tons applied in the center. This was the pressure required to bend the piece of $\frac{11}{16}$ of an inch.

The rod, however, showed no signs of a crack or even a tendency to rupture.

A rod of same dimensions made of $3\frac{1}{2}$ per cent nickel steel and heat treated was used in a similar test, and by applying a

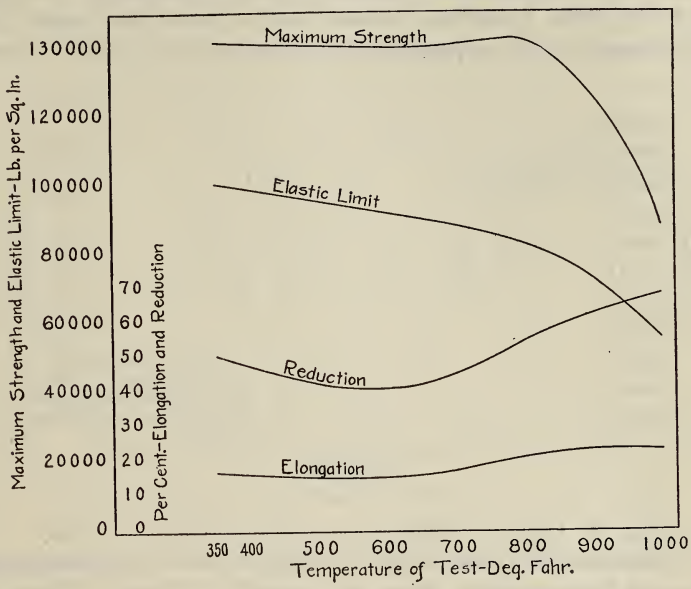


Fig. 64 Effect of Elevated Temperatures on Tensile Properties of Hy-Ten Steel

force of only 14 tons on the center of rod, it was found that the piece took a permanent set of $1\frac{1}{16}$ in.

This Hy-Ten steel is tough enough to be bent double upon itself under a hammer, without a fracture.

Fig. 64 shows the physical properties of this steel at elevated temperatures, and Figs. 65 and 66 the stress-strain curves.

Robert Cramer. Mr. Orrok did not fully point out the difference between the raising of temperature and pressure. The theoretical basis for high efficiency in steam engineering is the absorption of the maximum amount of heat at high temperature.

When pressures are raised, large quantities of heat are absorbed at high temperatures. When steam is superheated, only a small amount is absorbed at high temperature. Theoretically, the increase in pressure is more conducive to increased efficiency than the increase in superheat.

Mr. Orrok points out that at the present time it would pay the designing engineer more to investigate the possibilities in the refinement of the operation of plants, to remove such losses as are not due to the limitations of pressure and temperature. It is possible to improve a central station design from two angles, from the viewpoint of the operating practice and from the viewpoint

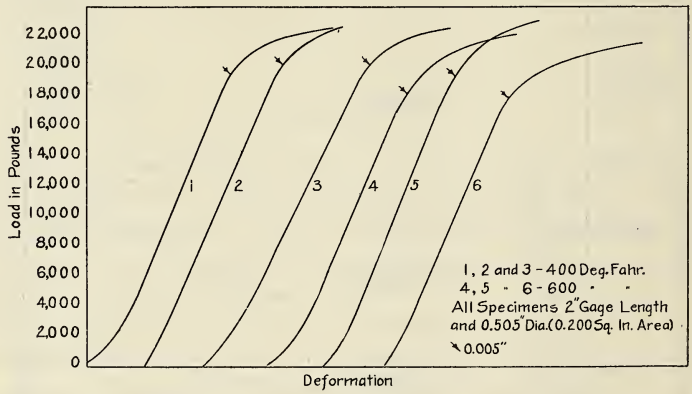


Fig. 65 Elevated Temperature Tests on Hy-Ten Steel

of the fundamental conditions of pressure and temperature, and the writer believes that these two should be kept distinct and separate.

Henry D. Hibbard. The matter of using the high temperatures in superheated steam containers contemplated by the author calls for far more knowledge than we now have of the behavior of steel and other metals when exposed to those temperatures for long periods of time, that is for years. Mr. Morrison gives a partial résumé of the properties at elevated temperatures and for a limited time of metals which would be employed in such work, but no one, so far as the writer knows, has investigated how the properties of such metals would be affected by long use in boilers for making highly superheated steam, and in apparatus for handling it in the way Mr. Orrok contemplates.

The first determinations of the properties of steel at elevated temperatures known to the writer were made by J. G. Howard at the testing laboratory of the Watertown Arsenal at Watertown, Massachusetts, and recorded in the Tests of Metals for 1888. Howard's work has been confirmed by later investigators, some of whom, however, failed to notice all the phenomena which he observed. When boiler steel is warmed it loses tensility (tensile strength) and gains ductility until its temperature is about 212 deg. fahr. which is a turning-point for those properties. The steel then has a maximum softness below 550 deg. fahr. As heating proceeds those properties take the opposite courses and the steel

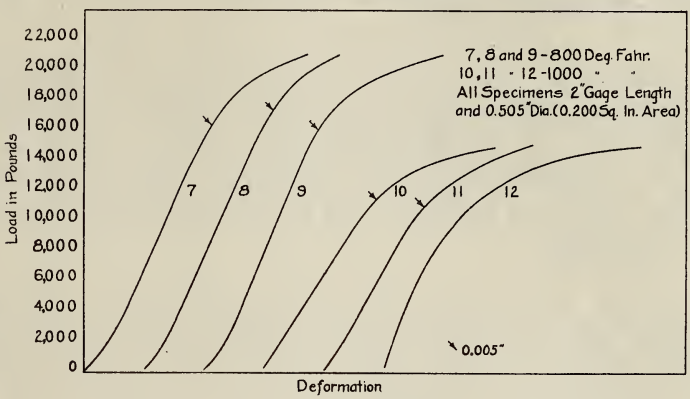


Fig. 66 Elevated Temperature Tests on Hy-Ten Steel

gains tensility and loses ductility up to about 550 deg. fahr. (the blue heat), where these tendencies reach a maximum and the metal is relatively unreliable. On further heating these tendencies are again reversed, tensility drops and ductility rises until the metal reaches the stage of incipient fusion where it loses its coherence. The elastic limit does not follow the tensile strength but falls somewhat regularly as the temperature rises from that of the atmosphere to the upper limit. What would happen to metals from long continued heating exposed to superheated steam at 550 deg. or the higher temperatures proposed by Mr. Orrok is a matter to be determined and that too before such conditions are adopted in practice. The recent discoveries relating to grain growth of metals have a bearing as, in a general way, steel loses some tensility and ductility as its grains grow larger which they tend to do at moderately elevated temperatures.

Howard found that cast gun-iron reached maximum tensility at 900 deg. fahr. or about 350 degrees above the temperature of maximum tensility of soft steel. This seems to be a favoring circumstance in the use of cast iron in handling superheated steam. The durability of andirons made of cast iron indicates that that iron under certain conditions may withstand the effect of long continued moderate heating better than wrought iron or steel. Perhaps the structure of cast iron interferes with grain growth.

Considering the temperatures employed in oil distillation as precedents in the matter is hardly warranted as the pressures are low in comparison with those contemplated by Mr. Orrok and superheated steam is absent.

Max Toltz. As a middle course between the high pressure and high temperature and the simplicity and ease of operation is the possibility of re-superheating steam between stages as was done in an experimental plant in Europe. With re-superheating between stages in a turbine it will not be necessary to exceed 650 deg. temperature.

Francis Hodgkinson said that manufacturers were embarrassed by lack of standards for pipes for the higher pressures and urged the Society to undertake the formation of such standards. He thought that the tendency to use alloy steel for bolts was unfortunate because of the danger of mixing such bolts with those of commercial steel when dismantling machinery.

Nevin E. Funk. The outstanding factors which must guide the design of a plant are, first, continuity of service, and then efficiency, flexibility, simplicity and cost, which are of about equal importance. Continuity of service and efficiency do not need explanation but perhaps flexibility does. By flexibility is meant a condition in which when something happens to a turbine, all the boilers supplying it are not put out of commission — all boilers must be available for any turbine. The major part of the plant must be capable of operation irrespective of individual units, and this flexibility must be obtained with as great simplicity as possible.

Oscar F. Junggren. The discussion of high steam pressure is not new. However, Mr. Orrok's paper gives a clear conception

of what may be expected due to an increase in pressure above what now is generally used, but such increase in pressure is not warranted unless it can be shown that it will pay.

In the past there has been a gradual increase in steam pressure. Each step has been accompanied by a development in the apparatus to suit the new condition. With the past experience before us it does not seem that the highest steam pressure has as yet been reached. Until someone gives the higher pressure a fair trial, we will never know what the efficiencies and the commercial possibilities will be.

H. B. Oatley. Mr. Orrok entirely disregards the influence of steam density upon the actual steam efficiency of the turbine. He assumes it to be 78 per cent in his calculations for a total temperature of 750 deg., and for a pressure range of between 200 and 1200 lb. Is this in accordance with the facts? It stands to reason that the average density of the steam throughout the turbine is greater in the case of an initial steam condition 1200 lb. and 750 deg. fahr. total temperature than with 200 lb. and 750 deg. fahr. total temperature. And is it not true that friction and windage losses in the turbine are a function of the steam density. The steam saving that is possible, with pressures now in use, through superheating, over and above the steam consumption with saturated steam, is in excess of the one indicated by a consideration of Rankine efficiencies alone. There is a distinct gain from superheat due to its decreasing the densities throughout the turbine stages, with the result that less heat of friction is imparted to the steam during its expansion, and the latter is more nearly adiabatic. All that makes it extremely probable that, as we increase the average density in the turbine, through use of higher pressures, the influence of the greater friction losses resulting therefrom will have to be compensated by substantial amounts of superheat.

Mr. Orrok states that the maximum temperature of the inner surface of a superheater tube may be a little higher than the maximum superheat, and that the metal temperature of the outer tube will depend upon the location of the superheater in the boiler, or, as he undoubtedly means, the gas temperature. There does not seem to be any very good reason why the temperature drop through the wall of a superheater tube made of steel should be any greater for the same heat transfer per square foot, and the same tube wall thickness, than it would be if the tube were filled with water. At

the same time, experimenters have found that the specific heat transfer from the inner tube wall of a superheater to the steam decreases with the density of the steam and steam velocity, and with increasing tube diameter. It would then seem that for a given total heat transfer, the temperature difference between the tube wall and the steam must increase as its density decreases, and as the diameter increases. It is therefore not possible that total steam temperatures of 1100 deg. can be produced, without considerably exceeding the steam temperature with the temperature of the tube metal. The only known means of holding down the excess of metal temperature are the employment of very much higher steam velocities and pressure drops than many engineers, in ignorance of the underlying facts, have been willing to concede to superheater designers, to use relatively small diameter tubes, and to restratify the steam often during the superheating process, with the object of thoroughly mixing up the denser core with the more rarefied concentric strata in contact with the tube wall, which move with a relatively low velocity and obstruct the heat transfer.

T. E. Keating. In seeking greater economy in steam power stations there is not a very great amount of improvement possible in the individual pieces of equipment, although whatever improvement in economy can be commercially justified should be demanded. Boilers and stokers are operating with efficiencies around 80 per cent and the greatest loss is in the flue gas, so that with the heat drop available between the furnace and boiler breechings, the actual combined boiler and furnace efficiency is better than 90 per cent. While turbine overall thermal efficiencies are less than 30 per cent due to the great loss of latent heat in the condenser circulating water, nevertheless on the basis of heat drop available for conversion into useful energy, modern large turbines have efficiencies better than 80 per cent, so that a very great amount of improvement in the turbine itself cannot be anticipated. Large generators are operating with efficiencies above 95 per cent, so they likewise leave very little opportunity for eliminating losses. Condenser circulating pumps, which constitute the greatest item for auxiliary power, also are designed with efficiencies in the neighborhood of 80 per cent. Forced- and induced-draft fans which are the next largest items do not have so high a degree of efficiency and present greater possibility for improvement.

Granting that the gains in individual apparatus, desirable as

they may be, must be relatively small, then improved thermal efficiency of steam stations must depend upon:

The use of this high-pressure range will require some new turbine the latent heat of part of the steam, thus reducing the losses to the condenser.

2 The use of heat reclaiming apparatus for eliminating losses in the flue gases

3 The utilization of a greater range of the heat cycle.

The first and third problems have been discussed in the excellent papers presented at this session,[1] and to complete the study there should be a paper discussing not only the thermal phases of air economizers for preheating the air of combustion, and water economizers for recovering heat to the feedwater, but also a discussion of the operating problems involved with such applications.

Referring to the paper by Messrs. Orrok and Morrison, and particularly to the appendix including test data on metals under high temperatures, it is evident that an increase of the temperature range is today more of a problem for the chemical and physical laboratories than for central stations, as modern stations have nearly reached the limits for materials now commercially applicable.

In his summary, Mr. Orrok states that pressures up to 1200–1500 lb. may be considered commercial and may be obtained without serious difficulty, and also that at present temperatures up to 700–750 deg. are commercially practicable. A statement, including some recommendations regarding the desirable combinations of these factors, i.e., pressure and temperature, would be of considerable interest. Assuming a steam station is to be built with commercially applicable equipment and a steam temperature of 750 degrees is chosen, would this value be obtained by the use of high pressure and a small degree of superheat or by the use of a lower steam pressure and a greater degree of superheat? The first condition would allow a greater available heat drop, which would be partly offset by the decreased efficiency of the high-pressure section of the turbine due to the steam leakage, and impaired efficiency in the low-pressure sections where the moisture content is high.

The use of this high-pressure range will require some new principles of turbine design if the value of 78 per cent turbine

[1] See also paper No. 1873.

efficiency used by the author is to be obtained in a single cylinder. If a compound turbine with reheating is used the cost of the additional equipment must be charged against the thermal gain from the higher pressure. The combined boiler and furnace efficiency will be lowered with the higher pressure, as the saturation pressure of steam at 1500 lb. is approximately 600 deg. fahr. Therefore either air or water economizers, or both, become a necessity and some portion of their cost must be charged against the gain in efficiency from the higher pressure. The values given in Fig. 2 are evidently based on an infinite number of stage heaters, although the author later stated that four or five stages have been so far the ultimate practical application.

While a continual increase of steam pressures will undoubtedly occur as in the past, it is the writer's feeling that when a practical system of feed heating is employed and proper consideration given to the increased costs of equipment, the net gain in commercial economy will be much less than might be anticipated, by merely considering the author's indicated thermal saving of 4000 B.t.u. per kw-hr. based on a nearly ideal cycle.

As soon as accurate data are available on the properties of steam at high pressures an estimate of the cost of power equipment for the pressures should be made with a study of the relative commercial economy of a station designed for 500 lb. and 280 deg. superheat and one designed for 1500 lb. and 150 deg., both of these conditions being equivalent to approximately 750 deg. total temperature.

H. LeRoy Whitney[1] said that there had been built and kept in operation for over two years a pipe line for gases at 1500 lb. pressure and 850 deg. fahr. temperature. At the largest point the diameter was 36 in. There was no trouble with steam piping from increased pressure, he said, if the temperature was kept below 700 deg. fahr.

E. H. Foster. The authors are to be warmly commended for both the title and the substance of this paper, dealing as it does with one of the most important problems now before the mechanical engineers.

Some six or seven years ago the writer became convinced of the soundness of this theory that steam engine economies might be

[1] M. W. Kellog Co., N. Y.

vastly improved by providing steam at higher pressure, and set out to build a water-tube boiler which would carry 500 to 600 lb. per sq. in. with 100 deg. superheat.

The boiler contained 2000 sq. ft. heating surface and was designed by the Winslow Safety Boiler Company. It comprised a number of vertical sections, each consisting of 2-in. steel tubes slightly inclined from the horizontal and attached at both ends to $2\frac{1}{2}$-in. vertical headers by welded joints. The sections were flexibly connected at the bottom to a mud drum and also at the top to a steam manifold which was only of sufficient diameter to avoid friction losses and was not intended to function as a steam drum. The upper part of the sections formed the steam storage and steam drying space, making in effect a cellular reservoir capable of withstanding excessive pressure without requiring thick walls, nor the corresponding weight of metal.

Superheating tubes integral with the sections were also provided and a novel form of economizer was designed for later installation.

The boiler was completed and operated for some time at a pressure around 500 lb. per sq. in. and gave, on test, an efficiency of over 70 per cent.

The steam was reduced to 125 lb. by a reducing valve, as required by the engines served.

The engine was designed by the Stumpf Unaflow Engine Company, Inc., and had two vertical, single-acting steam cylinders with trunk pistons 11 in. in diameter and 16 in. stroke.

The engine resembled a gas engine in appearance. The steam cylinder head only was jacketed. There were no cross-heads or guides, the connecting rod extending from a wrist-pin in the piston to the crank pin direct. The valves were of the poppet type, actuated by a cam on a lay shaft, which could be slid forward and back by an attachment to a lever, thus forming the simplest kind of a reversing motion. It was thought that this might be developed for marine engines, especially of the smaller sizes such as are found on tugboats or launches.

The entry of this country into the war, however, and the taking over of our works by the Government for the purpose of building superheaters for the emergency fleet, made it necessary to discontinue this work before the engine was erected, and finally the boiler material had to be scrapped before a trial of this promising com-

bination could be made. Nothing has developed since then to show that the principle on which we were working was incorrect, and I feel indebted to the authors for their further corroboration.

C. C. Trump. Mr. Orrok has ably opened up a subject in which I have been interested since 1912. Prof. J. Stumpf at that time showed me designs of a small una-flow engine which he said with proper high-pressure boiler and heaters would work on a regenerative cycle approaching a Carnot cycle in thermal efficiency. He did not know where to obtain a high-pressure boiler at that time, but this is now available through development by my associates, myself and others.

What Mr. Orrok says about large central stations with turbines also applies to small power plants with una-flow engines of correct design. These engines can be bled of steam for heating feedwater in stages and take even better advantage of a regenerative cycle than do small turbines. High pressures and temperatures have no terrors for una-flow engines with single-beat valves in steel cylinder heads. It is evident, however, that cast-iron heads and valves cannot be expected to survive higher temperatures than common at present. The data presented by Mr. Morrison are of special interest to engine designers for this reason.

Advantages of using well designed engines with high-pressure boilers may be summarized as follows:

1 Small space required
2 Exhaust steam available at any desired pressures
3 Fine efficiency on any fuel, solid as well as liquid
4 Low maintenance cost as compared with combustion engines
5 Safe and nearly automatic operation.

Geo. A. Orrok. In closing the discussion I again wish to emphasize the fact that in all figures above 400 deg. fahr, our fundamental data, the steam tables, are sadly lacking in accuracy. Nearly all values above this point are extrapolated and differences as large as 5 per cent are known to exist in published figures. Mr. Oatley says, "I disregard density of high-pressure steam in stating the efficiency of the Rankine cycle." I plead guilty but I believe that the lump figure of 78 per cent which I used will be below the actual figure that we will find when the high-pressure machines are tested and expressions from turbine designers have confirmed this opinion. Messrs. Keating and Wohlenberg have

commented on the cycles used, and criticized to some extent the steam generating efficiencies used. I have in all cases considered the boiler efficiency as boiler, furnace, stoker, economizer and air-heater efficiency, but I have not specified the auxiliaries used nor the means of attaining the regenerative cycle. Wohlenberg's tables are extremely interesting and show a number of solutions of the general problem.

Smoot's discussion of the principles of high-pressure turbine design follows out one of the main features of the program, the reduction of the heat rejected to the condenser, i.e., the making available of a larger portion of the latent heat. He also emphasizes the fact that the total heat of saturated steam passes through a maximum (although he does not say so) and the cost of heat in superheat becomes greater than the cost of heat in the liquid or in latent heat.

Dr. Jacobus has called attention to the work of Perkins in 1824 who worked with water at pressures above the critical point and apparently had little trouble with this portion of his apparatus. It is my understanding that Benson in England is making such proposals at the present time.

Junkersfeld's discussion is extremely valuable giving as it does certain prime-mover characteristics as well as a discussion of fixed charges per kw-hr. My use of 5000 hours per year was based on the average use of the best apparatus in power and lighting stations where the load factor never exceeds 50 per cent, the usual limits being 30 to 45 per cent.

Since the presentation of this paper actual propositions have been obtained from the manufacturers of all kinds of apparatus entering into a high-pressure station and it is believed that the figures given in the paper are considerably in excess of the true value.

Finally to improve our thermal efficiency we must cut down low level width or increase in the height of our entropy diagram. Increasing the height means high pressures, but we probably can go no higher in temperature for some time. Cutting down the low level width of the figure means less heat wasted to the condenser, a true saving of heat, a conservation of our fuel resources of considerable magnitude.

No. 1875

HIGH-TEMPERATURE AND HIGH-PRESSURE STEAM LINES

By B. N. Broido, New York, N. Y.
Member of the Society

The pronounced tendency of modern power-plant designers to use high superheat and high pressure, as well as the recent extensive use of highly superheated steam in process work, lead the author to make a review of the available data and formulas on radiation and friction losses in pipe lines. All tests to determine the radiation losses of pipe lines carrying high temperature steam made in this country have been practically exclusively with electrically heated pipes. This paper gives results and analysis of tests made with superheated steam flowing through a pipe line. The heat-transmission coefficients of bare and covered pipes at different steam velocities, various superheats, and pressures are given. Suggestions are made regarding steam velocities to be selected in order to transmit steam more economically.

The formulas available at present for figuring the friction losses of steam in pipe lines are limited to the range for which they were established, or for comparatively low pressures. Corrections of these formulas are shown in this paper, based on observations of a pipe line carrying high-pressure steam, which corrections may be used by designers of high-pressure power plants in dimensioning their pipe mains.

The paper further treats of pipe lines carrying superheated steam for other purposes than power. Recommendations as to the velocities or pipe sizes to be chosen are given, as well as a few examples from practice to show how those velocities were advantageously applied.

THE use of superheated steam in power plants, as well as for various kinds of process work, has become so extensive during recent years that it is hardly necessary to enlarge on its advantages. The fuel saving effected by the use of superheated steam, the low maintenance cost of turbines using it, and the numerous advantages it has in various kinds of process work are some of the factors which have helped to assign to it the important position it now occupies in the opinion of technical men.

2 One of the phases in which engineers, power-plant operators, and managers are naturally interested in this connection is that of

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conducting superheated steam through pipe lines. Very little, however, has been published in this country on the subject.

3 The recent tendency in designing of power plants, particularly those of large size, is toward high pressures. The high temperatures and high pressures involve new problems, among others being that of proportioning the pipe lines. A study of the losses of heat and pressure in pipe lines carrying highly superheated and high-pressure steam at the present opportune time will aid the designer in dimensioning the steam lines.

4 The field is very large, and this paper will therefore be confined to the discussion of —

a Radiation losses in pipe lines carrying superheated steam
b Resistance to the flow or pressure drop of superheated steam in pipe lines
c The most advantageous steam velocity, particularly for high-pressure lines; and
d Other features to be taken in consideration in designing pipe lines for transmission of superheated steam of either low or high pressures.

RADIATION LOSSES IN PIPE LINES CARRYING SUPERHEATED STEAM

5 Up to the present the usual method of determining radiation losses has consisted in supplying saturated steam to a pipe either naked or covered with a definite thickness of lagging, and weighing the amount of steam condensed. Such experiments, however, are mostly inaccurate, due to the fact that no separator completely removes the moisture from the steam, and also because of the difficulties in completely draining the tested pipe.

6 More accurate experiments to determine the radiation losses of pipes are conducted by heating the closed space inside the experimental pipe by means of electrical resistance coils. The product of the drop of voltage and the amperage gives a measure of the heat lost by the pipe.

$$\text{Volts} \times \text{amperes} \times 0.057 = \text{B. t. u. lost per minute from radiation}$$

7 So far as the author is aware, no experiments with superheated steam in pipe lines have been made in this country. Occasionally one finds in catalogs or other publications, data pertaining

to radiation loss with superheated steam. These data, however, were obtained with electrically heated pipes, and it was assumed that the radiation from a pipe carrying superheated steam of a certain temperature is the same as from a pipe heated up by an electrical resistance to the same temperature.

8 The author has had the opportunity to study this subject and to examine the results of tests conducted in Europe in which

TABLE 1 HEAT LOSS FROM 3-IN. AND 6-IN. BARE PIPES — SATURATED STEAM

Pipe Line								Heat Loss	
Length, ft.	Inside diam., in.	No. Flanges	Pressure abs., lb. per sq. in.	Steam temp., deg. fahr.	Air temp., deg. fahr.	Temp. diff. betw. steam air, deg. fahr.	Condensed water for 1 ft. length, lb. per hr.	Per sq. ft. surface per hr., B.t.u.	Per sq. ft. per deg. temp. diff. per hr., B.t.u.
87.25	2.76	6	46.60	274.46	60.98	213.48	0.5174	575.7	2.672
			46.01	273.56	59.54	214.02	0.5168	575.3	2.672
			46.45	274.10	61.88	212.22	0.5114	570.2	2.672
			96.43	322.52	65.66	256.86	0.6948	744.0	2.876
			96.87	322.88	62.96	259.92	0.7063	755.9	2.876
			95.84	322.16	70.34	251.82	0.6982	758.5	2.958
			190.22	374.72	76.10	298.62	0.9092	932.4	3.100
			191.69	375.26	71.06	304.20	0.9415	964.7	3.142
85.28	5.91	7	47.78	275.9	72.50	203.40	1.0362	545.4	2.672
			48.95	277.34	78.26	199.08	1.0087	537.2	2.652
			99.23	324.68	89.06	235.62	1.3628	689.3	2.897
			98.78	324.32	86.36	237.96	1.3803	698.9	2.917
			192.13	375.44	87.08	288.36	1.9320	935.7	3.223
			191.98	375.44	95.75	279.69	1.8715	906.9	3.223

actual superheated steam was used, particularly the tests made by Dr. Berner and Herr Eberle of Magdeburg and Munich, respectively.[1] Results and analyses of these tests are given briefly in the following paragraphs. Where advisable, the results with saturated steam are also shown for the sake of comparison.

9 The tests in Munich were conducted with a pipe line about 100 ft. long and about 3 in. (76 mm.) in outside diameter, one part of which — about 80 ft. in length — could be replaced by a pipe of about $6\frac{1}{4}$ in. outside diameter (108 mm.).

10 When the line was tested with saturated steam, all precautions were taken to remove any moisture in the steam before it

[1] Mitteilungen über Forschungsarbeiten (Berlin), V. D. I., Hefte 21 and 78.

entered the pipe line. In order to determine the radiation losses in a pipe line it is necessary to know the amount of steam condensed in the line. A few preliminary tests showed that with increased

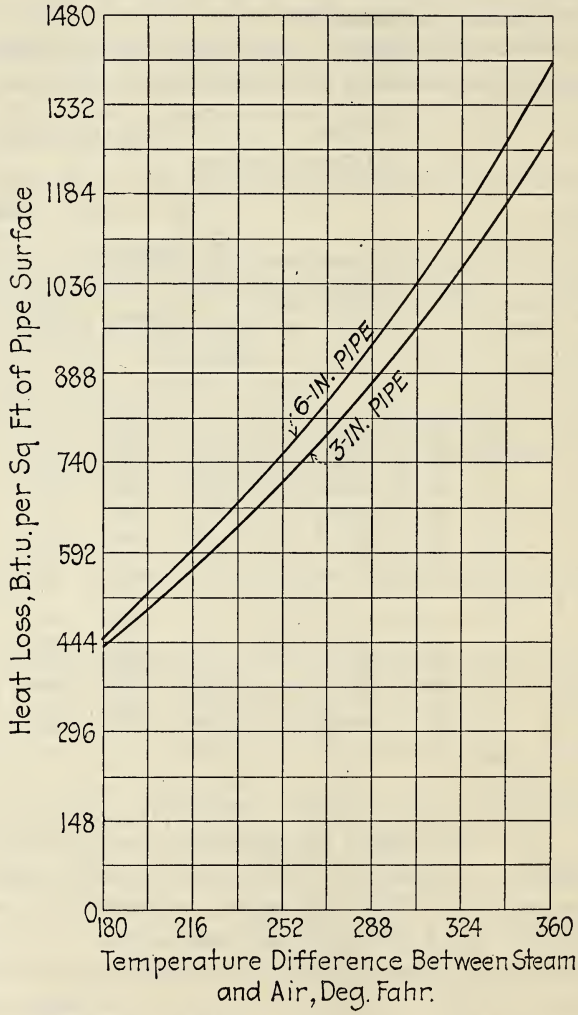

Fig. 1 Heat Losses from Bare Pipe — Saturated Steam

velocity of the steam the amount of water separated became low, which demonstrated conclusively that ordinary separators do not remove all the water from the steam, even at low velocities, and

that the higher the steam velocity, the less reliable the separators. It was therefore decided to conduct all saturated-steam tests by connecting one end of the line to the boiler and closing the other end, so that when the line was once filled with steam there would be no additional flow except to make up for condensation, and the steam would be practically at rest in the pipe. This could be done without influencing the results of the tests, as the wall temperature and the radiation losses are independent of the velocity of saturated steam. Special flanges were applied in order to collect all water condensed. Great care was exercised to keep conditions constant during each test. Readings were taken only after it was found that the pipe wall and the covering had reached and maintained their final temperature. With saturated steam it required from $1\frac{1}{2}$ to 2 hours to attain a steady temperature. With superheated steam, about four hours was required. With saturated steam, fairly constant conditions could be maintained during the test. However, with superheated steam fluctuations could not be avoided, due to the fact that it is much easier to maintain a constant pressure in a boiler than constant temperature or superheat in a superheater.

11 Table 1 shows the results of tests with saturated steam in bare pipes of approximately 3 in. and 6 in. internal diameter. The surfaces given in the table include the surface of the flanges.

12 In Fig. 1 curves are plotted showing how the radiation losses per square foot of pipe surface vary with the temperature difference between the steam and the outside air, while Fig. 2 shows how the coefficient of heat transfer per square foot of radiating surface per degree temperature difference per hour depends upon the steam temperature. It is an interesting fact that by inserting the values of Table 1 in the formula for the total heat-transmission coefficient K based on the well-known formula of Stefan-Boltzmann for the heat radiated from a black body, namely,

$$K = k - \frac{c}{t_w - t_a}\left\{\left(\frac{t_w + 460}{100}\right)^4 + \left(\frac{t_a + 460}{100}\right)^4\right\}$$

this coefficient corresponds very closely with that obtained by the results when the value of k, or the heat-transmission coefficient for convection, is taken as 1.2, which may reasonably be assumed to be correct. This shows that the Stefan-Boltzmann formula is reliable and can safely be used for similar calculations. In the above formula—

TABLE 2 CONDENSATION AND HEAT LOSS IN A 3-IN. COVERED PIPE LINE — SATURATED STEAM

Pipe Line			Pipe Covered with Flanges Exposed								
Length, ft.	Inside diam., in.	No. flanges	Pressure absolute, in lb. per sq. in.	Steam temp., deg. fahr.	Air temp., deg. fahr.	Temp. diff. between steam and air, deg. fahr.	Avg. temp. of covering, deg. fahr.	Outside surf. temp. of covering, deg. fahr.	Condensed water for 1-ft. length lb. per hr.	Heat Loss per sq. ft. surf. per hr. B.t.u.	Heat Loss per sq. ft. per deg. temp. diff. per hr. B.t.u.
87.25	2.76	6	46.45	274.46	64.04	210.42	154.04	85.82	0.1236	137.4	0.6528
			44.25	271.22	63.32	207.90	150.98	85.28	0.1210	135.3	0.6528
			97.90	323.60	69.44	254.15	178.52	96.80	0.1559	166.7	0.6528
			98.64	322.34	68.18	254.16	177.98	95.00	0.1599	170.0	0.6528
			191.69	375.26	75.02	300.24	201.02	107.42	0.1989	203.9	0.6732
			191.69	375.26	73.76	301.50	198.68	105.26	0.2043	209.4	0.6936
			Pipe and Flanges Covered								
87.25	2.76	6	45.13	272.48	52.34	220.14	145.94	74.12	0.0914	101.0	0.4692
			94.67	321.26	57.74	263.52	169.34	85.28	0.1142	122.5	0.4692
			95.26	321.81	57.56	264.25	169.16	84.56	0.1129	120.7	0.4488
			95.84	322.16	62.24	259.92	171.50	89.06	0.1129	120.7	0.4692
			191.98	375.44	71.96	303.48	199.40	106.34	0.1418	145.4	0.4692
			192.57	375.62	71.06	304.56	197.60	72.18	0.1378	141.2	0.4692
			190.66	374.90	64.58	310.32	192.92	97.52	0.1425	145.9	0.4692

K = total heat-transmission coefficient in B. t. u. per sq. ft. per hour per deg. fahr. temperature difference
k = heat-transmission coefficient for convection only
c = a constant, about 0.16
t_w = outside wall temperature, deg. fahr.
t_a = air temperature, deg. fahr.

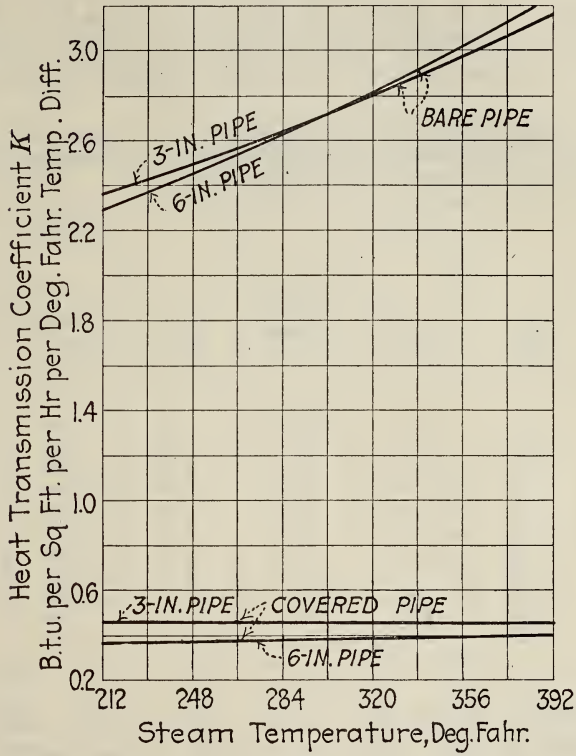

Fig. 2 Heat-Transmission Coefficient K of a Bare and Covered Pipe in B. T. U. per Sq. Ft. per Deg. Fahr. Temperature Difference per Hr., Depending upon the Steam Temperature — Saturated Steam

It is of interest that while the temperature of the pipe wall was very near to that of the steam, the temperature of the flanges was slightly lower. The difference in some cases is as high as 30 deg. fahr.

13 The tests were also conducted with different types of insulation, the results, however, being similar to those obtained in

like tests in this country. Table 2 shows results of experiments conducted with a covering consisting of 2-in.-thick shells of a porous burned material similar to kieselguhr, wrapped with canvas and painted twice with asphalt varnish.

14 Fig. 2 shows also the coefficient of heat transmission of the covered pipes.

15 Tests to determine the heat loss by transmission of superheated steam can be carried out only with steam that is flowing. The heat content of steam, both at the inlet and outlet of the pipe, must be determined, and the difference multiplied by the amount

TABLE 3 COEFFICIENTS OF HEAT-TRANSMISSION FOR BARE AND COVERED PIPES AT VARIOUS STEAM TEMPERATURES AND SAVINGS RESULTING FROM COVERING PIPE — SATURATED STEAM

Final steam temp., deg. fahr.	3-in. Line			6-in. Line		
	K		Saving through covering, per cent	K		Saving through covering, per cent
	Bare pipe, B.t.u.	Covered pipe, B.t.u.		Bare pipe, B.t.u.	Covered pipe, B.t.u.	
212	2.4	0.47	80.4	2.32	0.381	83.6
230	2.48	0.47	81.1	2.42	0.383	84.2
248	2.57	0.47	81.7	2.52	0.385	84.7
266	2.64	0.47	82.3	2.62	0.387	85.3
384	2.73	0.47	82.8	2.72	0.389	85.7
302	2.81	0.47	83.4	2.81	0.391	86.1
320	2.89	0.47	83.8	2.91	0.393	86.5
338	2.98	0.47	84.2	3.01	0.395	86.9
356	3.06	0.47	84.6	3.1	0.397	87.2
374	3.14	0.47	85.0	3.2	0.399	87.5
392	3.22	0.47	85.4	3.3	0.401	87.9

of steam flowing through the pipe gives the heat loss. Superheated steam flowing through a pipe may change its heat contents —

a By decreasing its temperature
b By pressure drop
c By partial condensation.

The accurate measurement of any condensed water would be impossible, due to the fact already mentioned that separators are not reliable. To be accurate, tests with superheated steam must therefore be conducted with sufficiently high temperatures so that the temperature of the pipe wall at the outlet does not descend below that of saturated steam at the same pressure.

16 Knowledge of the wall temperature was considered important, not only to determine the minimum superheat required to eliminate condensation, but also to determine the coefficient of heat transfer from superheated steam to a pipe wall. The measured temperatures of the pipe wall with steam at approximately 100 lb. pressure and at temperatures ranging from 200 to 300 deg. cent. (428 to 588 deg. fahr.), are given in Table 4.

TABLE 4 PIPE-WALL TEMPERATURE — SUPERHEATED STEAM

Pressure, abs., lb. per sq. in.	Final temp., deg. fahr.	Superheat, deg. fahr.	Air temp., deg. fahr.	K per sq. ft. per deg. temp. diff. per hr., B.t.u.	Wall temp., deg. fahr.	α_1 per sq. ft. per deg. temp. diff. per hr., B.t.u.	Steam velocity, ft. per sec.	Temp. diff. betw. steam and wall, deg. fahr.
				Covered Pipe				
93.8	540.8	217.6	50	0.70	530	31.8	68.3	10.8
92.5	549	226.8	50	0.72	540	39.8		9.0
93.8	551.2	228.0	50	0.72	544	49.8		7.2
				Bare Pipe				
93.8	556	232.8	51.8	3.79	493	30.4	98.5	63.0
96.7	567	230.6	51.8	3.85	513	36.7		54.0
98.2	579.8	253.3	51.8	3.9	524	36.7		55.8
95.3	586.2	261.9	51.8	3.9	525	34.0		61.2
95.3	588.2	263.9	51.8	3.9	527	34.0		61.2
95.3	588.2	263.9	51.8	3.9	527	34.0		61.2
95.3	586.2	261.9	51.8	3.9	525	34.0		61.2
98.2	513	186.5	51.8	3.61	403	15.2	29.5 to 32.8	110.0
96.7	495	169.6	51.8	3.55	396	15.8		99.0
95.3	476.2	151.9	51.8	3.49	388	16.8		88.2
95.3	464	139.7	51.8	3.38	363	15.2		101.0
95.3	440.8	116.5	51.8	3.32	358	15.6		82.8
95.3	442.4	118.1	51.8	3.30	356	15.6		86.4
95.3	434	109.7	51.8	3.28	353	15.6		81.0
96.	430.2	105.3	51.8	3.28	351	15.6		79.2
95.3	428.2	103.9	51.8	3.24	349	15.4		79.2

17 In view of the fact that the velocity of superheated steam doubtless has an influence upon the heat transmission, tests were conducted with different velocities. The results of tests conducted with bare pipes were highly impressive. At a velocity of about 100 ft. per sec. and steam temperatures of 291 to 390 deg. cent. (556 to 734 deg. fahr.), corresponding to 221 and 399 deg. superheat,

the temperature difference between the steam and pipe wall was 34 deg. cent. (61 deg. fahr.), while α_1 the heat-transmission coefficient was 35. With steam velocities of 33 ft. per sec. the difference

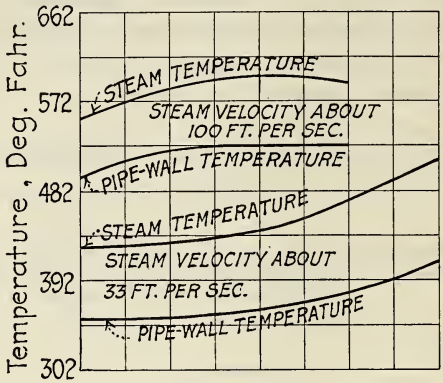

FIG. 3 INFLUENCE OF STEAM VELOCITY ON THE PIPE-WALL TEMPERATURE

between the steam and the pipe-wall temperatures was greater. At the lower steam temperatures of 220 to 267 deg. cent. (428 to 513 deg. fahr.) the temperature difference was 45 to 65 deg. cent. (86 to

TABLE 5 WALL TEMPERATURE OF BARE PIPE FIGURED FOR A HEAT-TRANSMISSION COEFFICIENT $\alpha_1 = 150$ — SUPERHEATED STEAM

Final steam temperature, deg. fahr.	Air temperature, deg. fahr.	Temperature difference between steam and air, B.t.u.	K, B.t.u.	Heat loss per square foot surface per hour, B.t.u.	Temperature difference between steam and pipe wall, deg. fahr.	Wall temperature, deg. fahr.
212	68	144	2.37	344	10.8	201
257	68	189	2.55	485	16.8	240
302	68	234	2.73	643	21.6	280
347	68	279	2.93	825	27.0	320
392	68	324	3.12	1021	32.4	360
437	68	369	3.30	1228	39.6	398
482	68	414	3.49	1454	46.8	436
527	68	459	3.67	1702	55.8	473
572	68	504	3.85	1961	63.0	509
617	68	549	4.04	2238	72.0	545
662	68	594	4.23	2538	82.8	580
707	68	639	4.42	2849	91.8	615
752	68	684	4.61	3078	102.6	649

122 deg. fahr.) and the heat-transmission coefficient, α_1 was 16. By decreasing the velocity of the steam one-third, the coefficient of heat transmission between superheated steam and the pipe wall was

decreased to one-half of the first value. This indicates clearly that the behavior of super-heated steam is different from that of saturated steam as far as imparting heat to a metallic wall is concerned, which is due to the fact that saturated steam, in transferring heat to the pipe wall, partly condenses or gives up some of its latent heat, which heat is not as closely combined with the steam as the sensible heat of a liquid or the heat of superheat. This fact often is not understood, even by technical men.

18 While the results of the tests with bare pipe — particularly

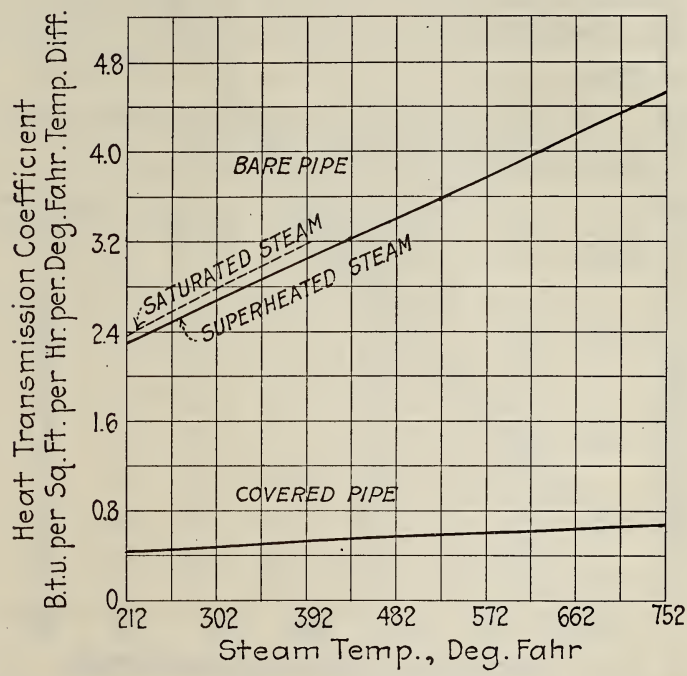

Fig. 4 Heat-Transmission Coefficient K of a Bare and Covered Pipe in B. T. U. per Sq. Ft. per Deg. Fahr. Temperature Difference per Hr., Depending upon the Steam Temperature — Superheated Steam

the variation of the heat-transmission coefficient at different temperatures and velocities — are not important in connection with pipe lines which invariably are insulated, they are of considerable interest as far as the use of superheated steam for heating and drying purposes is concerned.

19 The experiments demonstrate that the heat-transmission

coefficient increases considerably with the increased velocity, and also that the wall temperature depends not only upon the steam temperature but also upon its velocity. The results given in Table 4 are presented graphically in Fig. 3. The temperature drop between steam and the pipe wall, as given, applies only to one certain steam

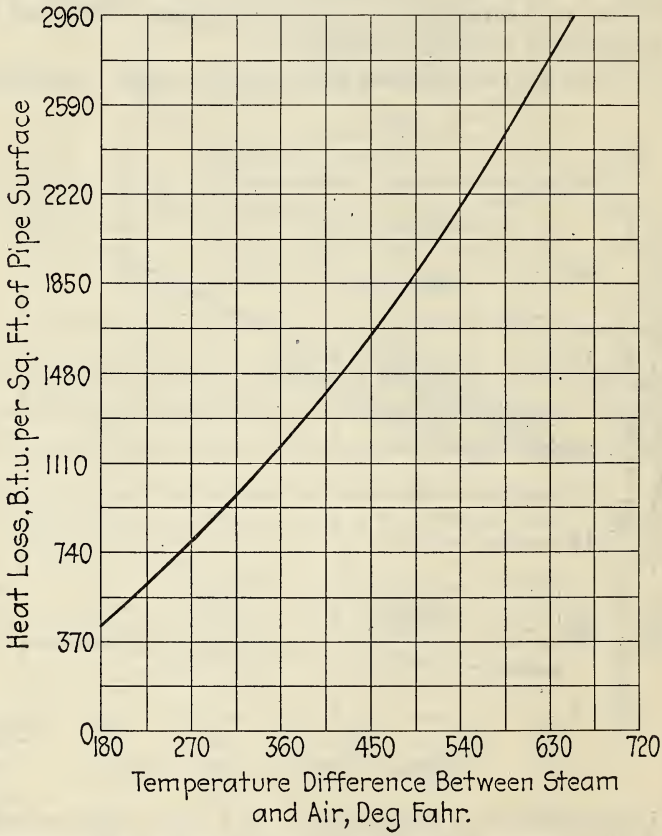

Fig. 5 Heat Loss from Bare Pipe — Superheated Steam

velocity. The wall temperature given in Table 5 corresponds to a heat-transmission coefficient of $\alpha_1 = 32$, corresponding to a steam velocity of about 83 ft. per sec. and a final temperature of 400 deg. cent. (752 deg. fahr.) While the table shows a wall temperature of 343 deg. cent. (649 deg. fahr.) at a steam temperature of 400 deg. cent. (752 deg. fahr.), the wall would have a temperature of only

314 deg. cent. (597 deg. fahr.) at a heat-transmission coefficient α_1 of 20.

20 While in the tests with saturated steam the steam temperature remained practically constant for the entire length of the pipe, when superheated steam is flowing through a pipe its temperature decreases. The temperature difference was taken into consideration in the values given. Table 4 gives the heat-transmission coefficients for bare and covered pipe of about 3 in. diameter, for steam temperatures from 220 to 291 deg. cent. (428 to 528 deg. fahr.), figured from the test results. Fig. 4 shows the same results graphically, while Fig. 5 gives the curve of the heat loss in B.t.u. per sq. ft. per hr.

TABLE 6 HEAT-TRANSMISSION COEFFICIENT K OF A 3-IN. (APPROX.) BARE PIPE — SUPERHEATED STEAM

Final steam temperature, deg. fahr.	Air temperature, deg. fahr.	Temperature difference between steam and air, deg. fahr.	K, B.t.u.	Heat loss per square foot surface per hour, B.t.u.
212	68	144	2.40	350
257	68	189	2.53	382
302	68	234	2.69	636
347	68	279	2.85	803
392	68	324	3.02	984
437	68	369	3.19	1191
482	68	414	3.38	1406
527	68	459	3.57	1650
572	68	504	3.77	1917
617	68	549	3.98	2200
662	68	594	4.20	2505
707	68	639	4.43	2849
752	68	684	4.69	3133

21 By inserting in the Stefan-Boltzmann formula the wall temperature in Table 5 corresponding to $\alpha_1 = 32$, the values given in Table 6 for the heat-transmission coefficient K, and also for the heat losses per hour, can be obtained by varying k, or the heat-transmission coefficient for convection, between 1.3 and 1.72, for temperatures from 100 to 400 deg. cent. (212 to 752 deg. fahr.). The increase of K with the higher temperature does not follow the same rule as that of saturated steam, as with the latter k remains unchanged. While the test results did not disclose any law governing the variation of k, there is no doubt but that it increases with increased temperature, particularly at the higher temperatures, which is

probably due to the fact that at the high temperatures the velocity of the air around the pipe also increases. In Fig. 4 the heat transmission coefficient for saturated steam is also shown by dotted lines. While the values are naturally higher than for superheated steam, it is, however, clearly seen that the slope of the curves is approximately the same in both cases, which is due to the fact that the radiation losses depend rather upon the wall temperature than upon the temperature of the steam, and shows again that the Stefan-

TABLE 7 VARIATION OF HEAT LOSSES, HEAT-TRANSMISSION COEFFICIENTS FOR BARE AND COVERED PIPES AND SAVINGS RESULTING FROM COVERING PIPE, WITH TEMPERATURE DIFFERENCE BETWEEN STEAM AND AIR — SUPERHEATED STEAM

Temperature difference between steam and air, deg. fahr.	Heat loss per square foot of outside pipe surface per hour		Saving through the covering, per cent	K	
	Bare pipe, B.t.u.	Covered pipe, B.t.u.		Bare pipe, B.t.u.	Covered pipe, B.t.u.
212	453	78.5	82.7	2.50	0.43
257	606	103.2	83.0	2.67	0.45
302	773	130.2	83.2	2.83	0.48
347	971	160.5	83.5	3.06	0.51
392	1184	192.4	83.7	3.26	0.53
437	1394	225.2	83.8	3.43	0.55
482	1637	260.8	84.1	3.63	0.57
527	1909	303.4	84.1	3.83	0.60
572	2192	344.1	84.3	4.04	0.63
617	2497	384.8	84.6	4.22	0.65
662	2804	429.2	84.7	4.43	0.67
707	3135	478.7	84.7	4.61	0.70
752	3149	529.8	84.8	4.81	0.73

Boltzmann formula is correct for calculating radiation losses from pipes.

22 Experiments were also conducted with a pipe line approximately 6 in. in diameter and with superheated steam. The results were entirely identical with those of the smaller pipe, which proves that the calculation is reliable and that the heat-radiation losses, contrary to the general belief, are little influenced by the diameter of the pipe.

23 Table 7 gives the results obtained with superheated steam in covered pipes. The same covering was used as for the test with saturated steam. A comparison of the values of Tables 3 and 7 for saturated and superheated steam in covered pipes brings out the fact

that for the steam-temperature range from 100 to 200 deg. cent. (212 to 392 deg. fahr.), with either saturated or superheated steam, the heat-transmission coefficient K remains practically the same. With increased temperatures this coefficient shows a tendency to rise considerably. For temperatures from 100 to 200 deg. cent. (212 to 392 deg. fahr.), K for lines with pipes and flanges covered is about 0.47. It increases, however, with the steam temperature, up to 0.69 at 400 deg. cent. (752 deg. fahr.).

24 In order to determine accurately the influence of temperature, velocity, or other varying conditions upon the losses, laboratory tests are usually conducted in enclosed rooms with the air practically still, while in actual practice the velocity of the air also has considerable influence upon the heat radiation. The tests showed that with the bare pipe the difference in temperature between superheated steam and the pipe wall is considerable. It is less, however, with covered pipes, and the more efficient the covering or the less the heat radiated from the pipe, the lower is the difference between steam and wall temperature. The higher the velocity of the air around the pipe, the more heat is radiated, and the greater the difference between the amount of heat radiated with saturated and with superheated steam. In practice the covering is seldom perfect, particularly that of the flanges, which are very often neglected. If the pipe-wall or flange temperatures are lower than the temperature of the steam, the radiation is naturally reduced. Observations of long outside pipe lines have shown that the actual radiation losses at an average air velocity of about 5 ft. per sec. are about 15 per cent to 30 per cent for saturated steam, and for superheated steam from 10 per cent to 20 per cent higher than the laboratory tests.

25 While from the tests one would conclude that the direct radiation losses for superheated steam are not appreciably lower than those of saturated steam of the same temperature, there are, however, a few other points to be considered in estimating the relative advantages of the two so far as their transmission through pipe lines is concerned. With saturated steam any heat radiated causes a part of the steam to be condensed, and the condensate, particularly for long pipe lines, is not returned to the boiler but is discharged through traps and wasted, so that in addition to the direct heat lost by radiation there is a further loss of the liquid heat of the condensate, which in some cases, particularly for high-pressure steam, may amount to 25 per cent of the radiated heat. With super-

heated steam the heat lost only decreases the temperature and no condensation occurs as long as the steam remains superheated, so that no additional heat besides that radiated is wasted.

26 Another important fact to be considered in connection with superheated steam is that a considerably higher velocity in the pipe lines is permissible. The velocity of the steam in a pipe, or the size of the pipe, is usually determined by the maximum friction loss or pressure drop which can be allowed.

27 Due to the low pressure drop and high velocity possible with superheated steam a greater amount of steam can be trans-

TABLE 8 VALUES OF HEAT-TRANSMISSION COEFFICIENT K FOR BARE AND COVERED PIPE WITH SUPERHEATED STEAM AND SAVING EFFECTED BY COVERING PIPE

Final steam temperature, deg. fahr.	K		Heat saved by covering, per cent
	For bare pipe, B.t.u.	For covered pipe, B.t.u.	
212	2.37	0.42	82.3
257	2.55	0.44	82.5
302	2.73	0.47	82.8
347	2.93	0.50	83.1
392	3.12	0.52	83.3
437	3.30	0.54	83.5
482	3.49	0.56	83.7
527	3.67	0.59	84.0
572	3.85	0.61	83.9
617	4.04	0.63	84.0
662	4.23	0.65	84.4
707	4.42	0.68	84.5
752	4.61	0.71	84.7

mitted with the same pipe, so that the radiation losses per unit weight are less. Therefore, in order to make a fair comparison between the radiation losses in carrying superheated and saturated steam, not the direct heat should be considered, but the total heat losses in percentage of the heat conveyed in the steam through the pipe. Table 9 gives the heat losses in pipe lines from 4 to 12 in. in diameter for saturated steam and for steam superheated 100, 150, and 200 deg., at gage pressures of 100, 150, and 200 lb. The values are calculated using the heat-transmission coefficients as given in Tables 3 and 7, and the values for saturated steam include the liquid heat of the condensed steam.

PRESSURE DROP OF SUPERHEATED STEAM IN PIPE LINES

28 The question of friction loss in pipe lines carrying gases or vapors has been studied by many authorities, and a number of formulas have been established which differ considerably. The most reliable tests, probably, are those carried out by Fritsche [1] with air. Some tests have also been conducted with superheated steam, and no appreciable difference in the friction was found. The latter tests, however, particularly those of Eberle, Biel, etc., have been conducted in short pipe lines provided with valves and bends — where most of the friction loss occurs — so that the difference, if any, between saturated and superheated steam could not be appreciable.

29 The author has had the opportunity of making observations on the friction loss in long pipes with both superheated and saturated steam, and after a considerable study of the subject, has come to the conclusion that, particularly for long pipe lines and moderate pressures, the formula —

$$P = \frac{C\,W^2L\left(1 + \frac{3.6}{d}\right)}{Vd^5}$$

which varies with the diameter of the pipe — using the coefficient 0.0001321 suggested by Babcock — is approximately correct for smaller pipe lines up to 4 in. in diameter; while for larger pipes, especially with wet steam, the formula $P = CW^2L/Vd^5$, with the coefficient suggested by Martin, Hawksley or Gutermuth, varying from 0.0003135 to 0.0003557 and depending upon the wetness of the steam and the surface of the pipe, is more likely to be correct.

30 For superheated steam with low temperatures and low velocities so that the wall temperature is lower than that of saturated steam and the walls are therefore covered inside with a film of water, the pressure drop is approximately the same as that of dry saturated steam. With higher temperatures, higher velocities and correspondingly higher wall temperatures so that the pipe remains dry inside, the coefficient C for superheated steam varies not only with the diameter of the pipe but also with the velocity, the pressure, the absolute temperature, and the density; and the friction loss agrees closely with that observed by Fritsche, so that his suggested formula for the coefficient C which follows can be considered reliable and used

[1] Mitteilungen über Forschungsarbeiten, V. D. I. (Berlin), Heft 60.

TABLE 9 COMPARISON OF CAPACITY, RADIATION LOSS, AND PRESSURE DROP OF PIPE LINES TRANSMITTING SATURATED AND SUPERHEATED STEAM

Saturated Steam								Superheated Steam								
4-In. Pipe (Extra Strong Above 125 Lb. Pressure)																
Press. abs., lb. per sq. in.	Press. drop, lb. per sq. in.	Velocity ft. per min.	Wt. of steam, lb. per hr.	Million B.t.u. trans. per hr.	Rad. loss	Total loss	Per Cent loss	Press. abs., lb. per sq. in.	Deg. superheat	Press. drop, lb. per sq. in.	Velocity, ft. per min.	Wt. of steam, lb. per hr.	Million B.t.u. trans. per hr.	Rad. loss	Per cent loss	Per cent saving
115	0.54	3000	4,080	4.12	16,060	19,400	0.47	115	100	1.24	6,000	7,020	7.46	26,400	0.34	27.7
									150	1.18	6,000	6,575	7.15	28,800	0.40	14.9
									200	1.12	6,000	6,225	6.92	33,200	0.48	.....
165	0.79	3000	5,170	5.25	17,650	22,180	0.42	165	100	1.75	6,000	8,975	9.65	26,850	0.28	33.3
									150	1.67	6,000	8,400	9.22	31,200	0.34	19.01
									200	1.58	6,000	7,910	8.90	35,800	0.40	4.76
215	1.03	3000	6,560	6.67	18,900	24,375	0.37	215	100	2.17	6,000	11,480	12.40	28,800	0.23	37.9
									150	2.04	6,000	10,750	11.78	33,400	0.28	24.3
									200	1.94	6,000	10,135	11.20	38,600	0.34	8.1
6-In Pipe (Extra Strong Above 125 Lb. Pressure)																
115	0.36	3000	9,660	9.75	23,650	28,780	0.29	115	100	0.75	6,000	16,000	17.66	37,400	0.21	27.6
									150	0.71	6,000	15,550	16.92	42,450	0.25	13.8
									200	0.67	6,000	14,710	16.35	49,000	0.30	.....
165	0.55	3000	10,520	10.78	26,000	32,670	0.30	165	100	1.04	6,000	20,280	21.80	39,400	0.18	40.0
									150	0.99	6,000	19,000	20.85	46,000	0.22	26.7
									200	0.94	6,000	17,900	20.15	52,650	0.26	13.3
215	0.72	3000	13,500	13.75	27,900	36,000	0.26	215	100	1.29	6,000	25,950	28.00	42,400	0.15	42.35
									150	1.22	6,000	24,320	26.85	48,900	0.18	30.8
									200	1.16	6,000	22,950	25.90	56,700	0.22	15.4

8-In. Pipe (Extra Strong Above 125 Lb. Pressure)

									100	0.71	7,000	32,410	34.5	48,600	0.14	33.3
115	0.37	3500	18,820	19.0	31,800	39,500	0.21	115	150	0.67	7,000	30,390	33.1	55,100	0.17	19.05
									200	0.64	7,000	28,710	31.85	63,700	0.20	4.76
									100	0.99	7,000	41,470	44.45	51,380	0.12	29.4
165	0.54	3500	24,200	24.55	33,900	42,600	0.17	165	150	0.94	7,000	38,810	42.6	59,770	0.14	15.62
									200	0.89	7,000	36,580	41.08	68,610	0.17	.....
									100	1.23	7,000	53,040	57.23	55,130	0.10	33.3
215	0.70	3500	31,150	31.7	36,300	46,800	0.15	215	150	1.16	7,000	49,680	54.90	63,740	0.12	20.0
									200	1.10	7,000	46,880	52.97	73,980	0.14	6.67

10-In. Pipe (Extra Strong Above 125 Lb. Pressure)

									100	0.69	8,000	58,200	61.87	60,760	0.10	28.5
115	0.38	4000	33,850	34.15	38,410	46,750	0.14	115	150	0.65	8,000	54,500	59.30	68,900	0.12	14.25
									200	0.63	8,000	51,500	57.16	79,530	0.14	.....
									100	0.95	8,000	77,500	83.10	63,670	0.08	33.3
165	0.56	4000	45,200	45.88	42,280	53,130	0.12	165	150	0.91	8,000	72,500	79.60	74,580	0.09	25.0
									200	0.86	8,000	68,400	76.80	85,620	0.10	6.65
									100	1.19	8,000	99,100	106.9	68,800	0.06	40.0
215	0.72	4000	58,200	59.30	45,320	58,460	0.10	215	150	1.13	8,000	92,800	102.5	79,530	0.08	20.0
									200	1.07	8,000	87,600	98.9	92,320	0.09	10.0

12-In. Pipe (Extra Strong Above 125 Lb. Pressure)

									100	0.87	10,000	104,200	110.8	72,070	0.07	22.3
115	0.50	5000	60,720	61.27	45,570	55,470	0.09	115	150	0.82	10,000	97,800	106.4	81,600	0.08	11.12
									200	0.78	10,000	92,500	101.8	94,340	0.09	.....
									100	1.18	10,000	140,900	151.0	76,050	0.05	37.5
165	0.72	5000	82,170	83.4	50,100	62,950	0.08	165	150	1.11	10,000	131,900	144.8	88,470	0.06	25.0
									200	1.06	10,000	124,100	139.4	101,560	0.07	12.5
									100	1.45	10,000	180,100	194.3	81,600	0.04	42.8
215	0.93	5000	105,800	107.8	53,650	69,220	0.07	215	150	1.38	10,000	168,950	186.7	94,340	0.05	28.6
									200	1.32	10,000	159,150	179.8	109,500	0.06	14.29

safely in calculating the pressure drop for long pipe lines carrying superheated steam and moderate pressures:

$$C = 0.0000022\ (R/144)^{0.148}\ (T/pw)^{0.148} d^{-0.269}$$

where —

T = absolute temperature, deg. fahr.
p = absolute pressure, lb. per sq. in.
w = velocity, ft. per sec.
d = pipe diameter, ft.
R = 85.7 for steam.

In order to facilitate the use of this formula a table was calculated giving for air (R = 53.34) the values of 1,000,000 C corresponding to values of the ratio T/pw for pipe diameters ranging from 1 in. to 48 in. This table is given in Marks' Mechanical Engineers' Handbook. When used for steam its values must be increased by 7 per cent on account of the different value of R used.

31 Of all the problems involved in connection with the use of high-pressure steam the question of its transmission has apparently been given the least attention. A great number of tests have been conducted to determine the friction losses in pipe lines carrying various fluids, such as water, air, and steam of *moderate pressures*, which tests were used to establish a number of formulas showing the interrelation of the friction losses, the density and velocity of the fluid, and the shape and roughness of the conductor, etc. The accuracy of these formulas, however, is limited to the range for which they were established. Outside of this range their use is frequently misleading.

32 As far as the author knows, all formulas for determining pressure drop, particularly for steam, are based on tests where the pressures in no case exceeded 300 lb. per sq. in. For higher pressures they should be used with caution. In his extensive study of this subject and search for a formula which would cover all conditions, Biel [1] found that the pressure drop caused by the resistance to flow of any fluid, for the same diameter of pipe, the same length, same temperature, and the same constant R, can be expressed by the equation $Pa = CW^n p^m$, in which the values of the exponents are $n = 1.852$, $m = 0.852$. Approximately the same values were found by Brabbée,[2] who made extensive tests, particularly to determine

[1] Mitteilungen über Forschungsarbeiten, Heft 44.
[2] Zeitschrift des Vereines deutscher Ingenieure, 1916.

the resistance, in pipe lines, of high and low pressure, water and air heating. He found, however, that the exponents vary with the roughness of the pipe. For steel pipe his exponent is 0.853.

33 The question arises as to how far the known formulas for the resistance of flow are applicable to high-pressure and high-temperature steam. The most elaborate tests ever undertaken with high-pressure steam for prime movers were carried out by Dr. Wilhelm Schmidt,[1] and extended over a number of years. The pipe lines, however, were too small (only about 4 in. in diameter) and too short to establish new formulas or even to verify those already existing for pressure drop in pipe lines.

34 The high-pressure steam plant of a chemical company in Griesheim, Germany, was found to be more suitable for the purpose. This plant consisted of two 500-hp. boilers of the Stirling type, built for a pressure of 30 atmospheres (440 lb. per sq. in.). While carrying out tests on the boiler the author had the opportunity to observe the pressure losses in the pipe line which was about 150 mm. (approximately 6 in.) in diameter and 380 ft. long. Tests with different pressures and velocities have shown that the variations of pressure drop at different velocities and initial pressures are smaller than would be expected if the foregoing formulas were correct also for very high pressures. The exponents obtained were smaller than those given above in the preceding paragraph. It is therefore the opinion of the author that the pressure drop in *straight lines* carrying superheated steam of 300 to 600 lb. pressure is from 10 per cent to 25 per cent less than that calculated by the use of the existing formulas for moderate pressures.

35 A comparison of the results of tests made at pressures between 250 and 440 lb. with those obtained by using the Fritsche formula bears out this statement. The pressure drop in fittings and bends was carefully determined by mercury U-tubes, and the resistance to the flow, which is usually expressed in lengths of straight pipe, was considerably higher than formerly supposed. As compared with the formula of Conrad Meier, given in the paper of D. E. Foster,[2] the values obtained were from 20 per cent to 35 per cent higher.

36 The foregoing would tend to show that most of the pressure drop in pipe lines takes place in the bends, fittings, and valves

[1] Zeitschrift des Vereines deutscher Ingenieure, 1921.

[2] Effect of Fittings on Flow of Fluids through Pipe Lines, Trans. Am. Soc. M. E., vol. 42, p. 647.

especially the latter, rather than in the straight pipes. In particular, sudden restriction of area and change in direction of flow affect the pressure of the steam. This has been verified by careful tests even with moderate pressures. The author recently had the opportunity to observe pressure differences in pipe lines in connection with superheaters, and an appreciable amount of pressure drop which was attributed to the superheater was found to occur in the non-return valves, the orifice of the flowmeter, etc., while the pressure drop in the superheater itself, which consisted of long, straight pipes without any area restriction, was very low as compared with the pressure drop through the fittings.

BEST VELOCITY OF STEAM FLOW FOR USE IN HIGH-PRESSURE LINES

37 It is therefore apparent that in pipe lines carrying high-pressure steam, provided they consist of long, straight pipe sections, high velocities are permissible, and for high pressures — say, from 300 to 500 lb. — 10,000 to 12,000 ft. per min. seems to be the most advisable for lines over 5 in. in diameter and with a continuous steam flow, as, for instance, when the steam is used in turbines. With reciprocating engines, naturally, lower velocities are to be chosen, depending upon the length of the line, the percentage of cut-off, and the number of revolutions of the engine. For high-pressure reciprocating engines located at a considerable distance from the steam generator it was found advisable to make that portion of the steam line near the engine of a larger diameter in order to minimize the fluctuation of the steam flow due to the intermittent steam demands of the engine, and to allow higher velocities by means of a correspondingly smaller pipe diameter, near the boiler.

38 If a line carrying high-pressure steam contains a number of bends, fittings, and valves, lower velocities than those just mentioned should be employed if excessive pressure drop is to be avoided.

39 The higher cost of the larger sizes necessitated by lower velocities, and the strength of the material required for higher pressures, are factors to be taken in consideration in determining the size of the line.

40 The table mentioned in Par. 30 giving values of 1,000,000 C is intended only for low pressures. Table 10 has therefore been devised to give values of the same quantity for pressures up to 600 lb. and any superheat. Table 11 gives the pressure drop and radiation

losses for pipes from 8 to 12 in. in diameter and for 300 to 400 lb. pressure.

41 The author would have it understood at this point that what has gone before is by no means to be considered as a comprehensive study of the question of the transmission of high-pressure

TABLE 10 VALUES OF 1,000,000 C

Diameter in inches	Values of the ratio T/Pw						
	0.008	0.01	0.015	0.02	0.03	0.04	0.05
4	1.35	1.38	1.43	1.46	1.53	1.60	1.65
5	1.23	1.29	1.33	1.37	1.44	1.50	1.55
6	1.16	1.21	1.26	1.30	1.37	1.43	1.48
8	1.10	1.14	1.18	1.22	1.27	1.33	1.37
10	1.03	1.07	1.11	1.14	1.19	1.25	1.29
12	1.00	1.03	1.07	1.10	1.14	1.19	1.23

steam. It is merely an expression of certain views in connection with these pipe lines, derived from the few tests available.

PIPE LINE CARRYING SUPERHEATED STEAM FOR OTHER PURPOSES THAN POWER GENERATION

42 The question of friction losses or pressure drop in a pipe line, particularly with superheated steam, is of importance only when the steam is used for generation of power. If it is used for heating, drying, or process work, only its temperature need be considered, the pressure being of no consequence. Higher velocities can therefore be chosen. Very often the pressure has to be reduced before the steam be used, and a pressure drop in the pipe line is even desirable. With saturated steam, however, it is necessary to limit the velocity. The usual velocities with saturated steam in commercial pipe lines are about 3000 to 6000 ft. per min. Any considerably higher velocity increases the danger of water hammer, particularly in long lines with a number of bends, fittings, and changes in diameter, and when the line cannot be completely and continuously drained. In such lines high velocities with saturated steam cause considerable vibration, especially when *much* moisture is carried with the steam, due to the effect at higher velocities of the greater weight of water at any change in direction of the flow.

43 With superheated steam, particularly at sufficiently high

temperatures, practically no moisture is present in the line, and nothing prevents the use of the highest speeds desired. Velocities of 12,000 to 15,000 (and more) ft. per min. can be and have been applied in larger pipe lines without any undesirable effects. In figuring the total radiation losses in Table 9, velocities of 3,000 to

TABLE 11 PRESSURE DROP AND RADIATION LOSSES IN PIPE LINES CARRYING HIGH-PRESSURE SUPERHEATED STEAM

Pressure Abs. lb.	Deg. Super Heat	Pressure Drop	Velocity ft. per min.	Wt. Steam lb. per hr.	Million B.t.u. trans per hr.	Rad. loss	Pressure abs. lb.	Deg. Super heat	Pressure drop	Velocity ft. per min.	Wt. Steam lb. per hr.	Million B.t.u. trans per hr.	Rad. loss
4-in. Pipe (Extra Strong)													
315	100	2.60	6000	16450	17.92	32100	415	100	3.03	6000	21190	23.25	34750
	150	2.46	6000	15400	17.18	37150		150	2.84	6000	19740	22.20	39550
	200	2.34	6000	14470	16.50	42200		200	2.68	6000	18460	21.20	45250
6-in. Pipe (Extra Strong)													
315	100	1.58	6000	37300	40.60	47300	415	100	1.79	6000	47900	52.60	51350
	150	1.48	6000	34900	38.90	54750		150	1.68	6000	44600	50.20	58400
	200	1.39	6000	32700	37.40	62000		200	1.59	6000	41750	48.00	66900
8-in. Pipe (Extra Strong)													
315	100	1.44	7000	76000	82.80	60600	415	100	1.68	7000	97900	107.3	65750
	150	1.36	7000	71200	79.50	70200		150	1.57	7000	91250	102.6	75000
	200	1.29	7000	66900	76.40	79700		200	1.49	7000	85400	98.2	86000
10-in. Pipe (Extra Strong)													
315	100	1.37	8000	142100	154.8	76800	415	100	1.57	8000	183000	205.0	83250
	150	1.29	8000	133000	148.2	88700		150	1.48	8000	170400	192.0	94900
	200	1.22	8000	125000	142.7	100900		200	1.4	8000	159500	183.6	108700
12-in. Pipe (Extra Strong)													
315	100	1.65	10000	258500	282.0	91300	415	100	2.04	10000	332500	365.0	99000
	150	1.56	10000	242000	270.0	105200		150	1.90	10000	309500	348.0	112700
	200	1.47	10000	227000	259.0	119800		200	1.79	10000	290000	334.0	129200

5,000 ft. per min. were assumed for saturated steam, and 6,000 to 10,000 ft., for superheated steam, which are conservative values that enable a fair comparison to be made.

44 Table 9 shows that the greatest saving, as far as radiation losses are concerned, is obtained with 100 to 150 deg. superheat, so

that when the steam is superheated only for the purpose of elimination of condensation or reduction of radiation, a moderate superheat of 100 to 150 deg., depending upon the length of the line, is sufficient. It does not mean, however, that moderate superheat is in general more desirable. If the steam is used for power purposes, the advantages of high superheat in the prime mover will more than overbalance the slightly higher radiation losses.

45 A remarkable feature is the very slight increase of pressure drop, which throughout the whole range of the table, with one exception, does not exceed 1 lb. for 100 ft. of pipe length, in spite of the fact that the velocity was doubled. The lower density of superheated steam, the absence of moisture, the dry pipe walls, and the reduced amount of foreign matter — the greatest part of which is usually left in the superheater — account for the low pressure drop.

46 It is commonly believed that even with superheated steam the pipe walls are covered on the inside with a film of moisture. This is true for low superheat with low steam velocities. At higher superheat and high velocities, however, the wall attains a temperature higher than that of saturated steam, and remains dry. The author has made a study of the wall temperatures of apparatus carrying steam, particularly in connection with the use of superheated steam in radiators for drying or heating purposes, where it is of considerable importance, whether the inside surface is dry or wet. This, however, may be made the subject of a separate paper at a future meeting.

EXAMPLE SHOWING ADVANTAGES OF USING HIGHER VELOCITIES WITH SUPERHEATED STEAM

47 The three following examples from actual practice demonstrate how the possibility of applying higher velocities with superheated steam may be advantageously utilized in reducing the radiation losses and the cost of the piping and covering.

48 The management of a large textile concern, realizing that it was necessary to lower their power cost, decided as one means to this end to avail themselves of the advantages of superheated steam. Each of the five boilers in the plant was accordingly equipped with superheaters for 200 deg. superheat.

49 The steam was for use in two engines, one triple expansion and one compound. The compound engine was located about 950 ft. from the boiler house. The piping between boiler and engine

was entirely changed, the old 8-in. piping, being replaced by a new 6-in. line. With the old piping and saturated steam the average pressure drop in the line between boiler and engine was 6 lb. With the new line and superheated steam it was 7.5 lb. The comparatively low pressure drop in the new line was mainly due to the appreciably decreased steam consumption of the engine. The radiation loss of the 8-in. line would naturally be 33 per cent higher than that of the 6-in. line for the same steam temperature.

50 A chemical company manufacturing aniline dyes had an 8-in. pipe line, 1400 ft. long, in their plant, conveying steam from their boiler room to the process house. An increase in production required the approximate doubling of the steam consumption in the power house, and a second pipe line accordingly was planned. At the same time the use of higher temperatures was found advisable, since the boiler pressure could not be increased appreciably. Superheaters were installed in the boilers, and approximately double the amount of steam superheated was sent through the pipe. Pressure drop was of no consequence, while the radiation loss per pound of steam conveyed was decreased 36 per cent.

51 A heating plant had two mains 10 in. and 8 in. in diameter and 2300 ft. long. The 10-in. line was used in the cold winter months, while the smaller one was utilized during the remainder of the year. With saturated steam the larger steam demand in the winter months could not be supplied with the small pipe alone, due to the danger of water hammer and excessive vibration, and the larger pipe had to be used. In order to have dry steam at the end of the line, and avoid the losses in liquid heat of the steam condensed in the line, superheaters were installed, and it was found that the greater amount of steam could be conveyed through the smaller pipe without any difficulty. The radiation loss, as compared with the 10-in. line, was 20 per cent less.

SUMMARY

52 To summarize:

a The heat-radiation losses in B.t.u. per square foot of surface of an uncovered pipe carrying *saturated steam* are independent of the velocity of the steam.

b The temperature of the pipe wall is slightly lower than that of the saturated steam in a bare pipe, and approaches close to that temperature in a covered pipe.

c The velocity of *superheated steam* has considerable influence upon the heat loss by radiation of a bare pipe.

d The temperature of the pipe wall of a bare pipe carrying *superheated* steam is appreciably lower than that of the steam. The higher the temperature and the lower the velocity of the steam, the greater is the difference. With a covered pipe the difference is considerably less, and the more efficient the insulation, the smaller this difference.

e The heat-transmission coefficient between *saturated* steam and a pipe wall is approximately 400, and varies slightly only with the temperature difference between the steam and the outside air.

f With *superheated* steam the coefficient α_1 depends largely upon the velocity of steam. It was found to be about 32 at a steam velocity of 83 ft. per sec. and a steam temperature of approximately 750 deg. fahr., and to decrease rapidly with decreased velocity. Whether the pipe is dry or covered with a film of moisture also influences the heat-transmission coefficient.

g The coefficient of heat transmission from steam — either saturated or superheated — to the surrounding air remains practically constant up to a temperature of about 350 deg. fahr. and tends to increase at higher temperatures.

h When superheated steam is flowing through a pipe at a sufficiently high velocity and temperature so that the pipe remains dry inside, the friction loss is considerably less than with saturated steam of the same density and with the same velocity. With superheated steam higher velocities and smaller pipe sizes can therefore be chosen, also appreciably decreasing the radiation loss as well as the cost of the pipe line and covering.

i The values of the pressure drop in straight pipes, obtained by using existing formulas seem to be too high for pressures above 300 lb., and the error apparently increases with higher pressures.

j The known values of the lengths of pipes in which the resistance to the steam flow is equal to those of fittings or valves, are apparently too small for high-pressure steam, particularly with respect to valves.

k If superheated steam is used for other purposes than for power, and pressure drop is of no consequence, it can be conveyed in pipes at a very high velocity, as the danger from water hammer and vibration is eliminated.

DISCUSSION

E. G. Bailey. The factors considered in the design of steam lines have usually been limited to the cost of pipe and covering, the pressure loss, and the radiation loss. Proper conditions for the installation of steam meters should also receive due consideration.

The steam meter has now become such an integral part of the plant equipment for measuring the output of boilers, steam consumption of turbines and various units as well as the steam distribution, that the conditions suitable for the installation of such meters should be considered in the design of piping. This applies not only to the selection of the proper size of pipe but also to the layout for flow conditions as well as for proper expansion.

As a rule the piping is laid out with little or no consideration for meters and the meter manufacturer is later confronted with many difficult problems in finding a suitable location in which the primary element of a steam meter can be installed.

Practically all steam meters have a primary device such as the pitot tube, orifice, flow nozzle or venturi tube that is placed in the line to obtain a differential pressure. The most used primary devices for steam are the orifice and flow nozzle, which are inserted between a pair of flanges and produce a small pressure loss in the steam line. It is natural that this pressure loss should be held at a minimum and most meters operate on approximately a two-pound differential pressure, part of which is restored so that the actual pressure loss is in the neighborhood of one pound.

The length of straight run of pipe required on the inlet and outlet side of such orifice or flow nozzle is a question that cannot be answered without knowing the nature of the pipe or fitting on the beginning of the straight run and also the steam velocity or rather the kinetic energy with which it flows through the pipe. Steam flowing through elbows, globe valves, angle valves, Y-fittings and tees has more disturbance than when it comes through a gate valve or around a long radius bend and, therefore, a longer straight run is required in order that the stream lines may be normal by the time they approach the primary metering device. On the other hand if the velocity of steam is low an orifice can be placed very close beyond fittings or bends.

The change in energy due to the differential pressure across the primary metering device should be definitely related to the total

kinetic energy of the steam as it approaches the primary device in order that the desired accuracy be obtained.

Meters can of course be designed to operate on a greater differential pressure so as to maintain the ratio of energy suitable for accuracy. However, there will be a greater pressure loss in the line than most engineers desire, and it, therefore, comes back to a matter of giving proper consideration to the value of steam meters and their requirements in laying out piping or at least the section of piping in which the meter is installed.

The maximum permissible velocity for any given design of steam meter and its primary device is represented by the formula

$$\text{Velocity} = F\sqrt{\text{Specific Volume}}$$

where velocity is measured in feet per minute and F is a factor. This factor is not only dependent upon the meter construction but it also varies with the pipe design on either side of the orifice or primary device.

One meter manufacturer gives limiting velocities of $V_{max} = 8000\sqrt{\text{sp. vol.}}$ and another $V_{max} = 10{,}500\sqrt{\text{sp. vol.}}$ With steam at 250 lb. pressure and 200 deg. fahr. superheat having a specific volume of 2.35 the maximum velocities would be 12,240 and 16,000 ft. per minute respectively.

These velocities are capable of being metered accurately only when the steam is flowing through straight pipe of reasonable length on either side of the primary device.

Table 12 gives an approximate idea of the factors for different lengths of straight pipe and the effect of fittings.

L. B. McMillan. The writer is in full agreement with the author regarding the merits of superheated steam in general, but feels that the author's statements regarding relative rates of heat transmission from saturated and superheated steam to pipe walls are likely to be misinterpreted.

The author quotes the German investigator Eberle as authority for relatively low values of heat transmission from superheated steam to pipe wall, as compared with high values for saturated steam. It is a matter of common knowledge that heat is transmitted much more readily from saturated steam to a surface than from superheated steam. Eberle's data have also been available and have been already referred to by most investigators on heat transfer during the past ten years, although so far as the writer is

aware Eberle's paper has not previously been translated into English, even in part.

However, granting a large difference in transfer coefficients from saturated and superheated steam to pipe wall, the question is, how much difference will this make in actual heat losses.

TABLE 12 LIMITING CONDITIONS FOR MEASURED STEAM FLOW

Velocity of steam at sp. vol. 2.35, ft. per min.	Factor F	Pressure loss caused by meter at full capacity, lb. per sq. in.	Pipe condition required			
			Inlet side		Outlet side	
			Straight run, pipe diam.	Limit in style of fittings preceding straight run	Straight run, pipe diam.	Limit in style of fittings following straight run
16,600	10,850	2.0	15	No limit	5	No limit
			10	No Y fittings		
			6	No fittings except gate valve or long radius bend		
13,900	9,080	2.0	10	No limit	5	No limit
			8	No Y fittings nor globe valves		
			5	No fittings except gate valves and long radius bends		
10,100	6,600	2.0	8	No limit	5	No limit
			5	No Y fittings nor globe valves		
9,520	6,220	0.5	15*	No limit	1 or gate valve O. K.	No elbow nor globe valves
			10*	No Y fittings nor globe valve		
			5*	No fittings except gate valves		
6,150	4,020	0.8	8*	No limit	1 or gate valve O. K.	No elbow nor globe valves
			5*	No Y fitting nor globe valve		
			1	No fitting		
4,226	2,760	1.0	5*	No limit	1 or gate valve O. K.	No elbow nor globe valves
			1	No fitting		
1,900	1,240	1.5	2	No limit	No limit O. K. to go into any valve, elbow or fitting	No limit
			1	No Y fitting nor globe valve		

* May include bend if radius is 5 pipe diameters or more, and at least one pipe diameter of straight pipe on end of bend adjacent to flange.

Using for purpose of illustration the values of α given by the author, we have

α_1 = rate of heat transmission from superheated steam to pipe wall
= 32 B.t.u. per sq. ft. per deg. temperature difference per hr.

α_2 = rate of heat transmission from saturated steam to pipe wall = 400 B.t.u. per sq. ft. per deg. temperature difference per hr.

The value of α for saturated steam is thus over twelve times that for superheated steam and it would be only natural to expect that the loss from the saturated steam pipe would be far larger than that from a superheated steam pipe at the same temperature. Such would be the case if the transmission of heat from steam to pipe wall were the only factor involved, but the heat must be transmitted from the outside of the pipe to the air, or through insulation to the air. Therefore, the additional resistance to heat flow is so great outside of the pipe that the difference in inside resistances becomes very small in comparison.

This is illustrated as follows:

The inside surface resistance corresponding to $\alpha_1 = 32$ is

$$\frac{1}{32} = 0.031$$

and the inside surface resistance corresponding to $\alpha_2 = 400$ is

$$\frac{1}{400} = 0.0025$$

but for an efficiently insulated line a fair average resistance offered by the type and thickness of insulation ordinarily used on superheated steam lines in this country would be about 4.5. Therefore, the total resistances would be $4.5 + 0.031 = 4.531$ for superheated line, and $4.5 + 0.0025 = 4.5025$ for saturated steam line.

The actual heat losses are the reciprocals of these resistances, or $\frac{1}{4.531} = 0.221$ B.t.u. per sq. ft. per deg. temperature difference per hr. for superheated steam line, and $\frac{1}{4.5025} = 0.222$ B.t.u. per sq. ft. per deg. temperature difference per hr. for saturated steam line.

Note that these differ by less than one per cent.

In the case of bare pipes, there are larger differences, but even with bare pipes the differences are of the order of only 10 per cent. These differences are of little importance as few bare pipes are used for the transmission of superheated steam.

Most published tables on heat losses from bare pipes are based on loss per square foot per degree temperature difference between *pipe surface* and air. These are therefore correct for the stated condition regardless of what is inside the pipe.

An analogy illustrating how a big difference in resistances may make little difference in flow is as follows:

Suppose we have two pipes of equal length, one 2 in. diameter and the other 6 in. diameter, and both connected to the same water main. But suppose also that each is bushed down to 1/8 in. and an equal length of 1/8-in. pipe is added to each. The resistance to flow in the 2-in. pipe is many times that in the 6-in., but the flow will be practically the same in both cases because in either case the resistance of the smaller pipe is so great that the resistance of the larger pipe is not sufficient to affect appreciably the total.

Fig. 4 illustrates how small are the differences in losses from superheated and saturated steam pipes. Even in the case of the bare pipes, the difference shown by this test is less than 5 per cent and on the insulated pipes the results from actual test showed a slightly lower average loss on saturated than on superheated steam pipes. This is the opposite of what one would expect, but in any case the difference one way or the other on insulated pipes is probably within the limits of experimental error and consequently not worthy of consideration.

In Par. 25, the author concludes "that the direct radiation losses for superheated steam are not appreciably lower than for saturated steam at the same temperature," and turns to other advantages of superheated steam.

This conclusion seems to be entirely in accordance with the facts, but in view of it one wonders why the author devotes the major portion of his paper to presenting Eberle's data, apparently intended to show that there should be a considerable difference in losses and finally concludes that the difference is not appreciable.

Referring to Table 9 in which a lower radiation loss per pound of steam is shown for superheated steam than for saturated steam, this result is based entirely upon the assumption that the velocity in superheated steam lines is exactly double the velocity in saturated steam lines. This table shows higher actual heat losses on the superheated steam lines, but on the basis of the doubled velocity so much more steam is passed through the superheated steam line that the loss per pound is shown to be less. An explanation of just why the author assumed a doubled velocity in the superheated steam line instead of some other ratio would be necessary in order to give significance to this table.

In fact, it is the writer's understanding that the practice of calculating steam line sizes on an assumed velocity is antiquated

and that the preferred method of calculating steam line sizes is based on allowable pressure drop.

The foregoing discussion is in no sense opposed to the use of superheated steam, but is intended to show the relative magnitudes of the factors which determine losses and to emphasize the fact that in a problem of this kind consideration of only one of the contributing factors does not lead to conclusive results.

L. L. Barrett. Those engineers who have occasion to consider the heat losses from pipe lines conveying steam will be grateful to the author for his presentation of extracts from the paper of Chrs. Eberle.[1] The author's Figs. 1, 2, 3, 4, and 5 are identical with Eberle's Figs. 22, 31, 35, 37, and 38 respectively, and the author's Tables 1, 2, 3, 4, 5, 6, 7, and 8 are the same as Eberle's Tables 11, 13, 17, 25, 26, 27, 29, and 30 respectively, after converting from metric to English measures. Eberle's work has never been translated into English and readers have been obliged to refer to it in the original German in the past. It could be wished that more of it had been presented, as engineers will be interested as to what observations were taken and what measuring instruments were used. Tables 9 and 11 represent original computations based on Eberle's and Berner's[2] data and must therefore be considered in the light of such data.

Inasmuch as Berner's paper was published in Germany in 1905 and Eberle's paper was published in 1909, it is desirable that these data be examined to see how they compare with the more recent experimental work in this line in this country. Heat transmission data for pipes have been determined and presented to this Society by McMillan[3] in 1915, Bagley[4] in 1918, and Heilman[5] in 1922. These data are expressed as a function of the temperature difference between the surface of the pipe and that of still air. It is fundamental that the heat loss from a pipe to still air remains the same as long as this temperature difference is the same and irrespective of what the pipe contains. In these experiments the walls of the pipes were heated electrically, which enabled the very accurate measurement of the heat.

These tests were made with extreme accuracy and are in sub-

[1] *Mitt. über Forschungs-Arbeiten,* heft 78.
[2] *Idem,* heft 21.
[3] Trans. A.S.M.E., vol. 37, p. 968.
[4] *Idem,* vol. 40, p. 667.
[5] This volume of Trans., paper No. 1848.

stantial accord. In applying the data to pipe lines carrying saturated steam, the temperature of the pipe surface is assumed to be the same as that of the steam, the temperature drop from steam to pipe being less than 1 deg. fahr. for covered pipe. In applying the data to insulated lines containing superheated steam, the temperature of the pipe is taken at not more than 10 deg. fahr. less than the temperature of the steam. The correctness of this practice is confirmed by Eberle's results given in Table 4. The conveying of superheated steam in uninsulated pipe lines being practically unknown in this country, no experiments have ever been undertaken here to ascertain the temperature drop from the steam to the pipe in this case, and Eberle's results given in Table 4 constitute the only known data.

Referring now to the tables incorporated in this paper, the heat losses on which Tables 1, 2, and 3 are based were obtained by weighing the water condensed in the pipe line under test. This method is inaccurate, as stated by the author, and the later determinations of McMillan, Bagley, and Heilman are preferable. The constancy of the value of K for the covered 3-in. line in Table 3 is a good illustration of the inaccuracy of this table. Both McMillan and Bagley have shown that this value will vary considerably with the temperature difference.

The essential data in Table 4 are discordant. α_1 is the coefficient of heat transfer from steam to pipe wall in B.t.u. per sq. ft. per hr., per deg. fahr. temperature difference between steam and pipe. When the mean value of α_1 corresponding to each of the three steam velocities used is plotted as a function of the velocity it is to be expected that a curve could be drawn through the plotted points which would show an increase in the value of the transmission coefficient with an increase in velocity, it being remembered that this transmission coefficient from the steam to the pipe wall is entirely independent of what is outside the wall, i.e., whether the pipe is covered or not. But the three points do not lie in any reasonable form of curve which makes it very doubtful as to what value of α_1 to use for any required steam velocity.

Now it is well known, from the experiments of the Babcock & Wilcox Co.[1] and those of Fessenden and Haney,[2] that the value of α_1 increases not only with the velocity but also with some power of the absolute temperature of the hot gas. The Babcock & Wilcox

[1] Experiments on the Rate of Heat Transfer, B. & W. Co., 1916.

[2] Univ. of Mo. Bull. vol. 17, No. 26.

data indicate that α_1 varies as the one-third power of the absolute temperature while the Fessenden data follow the two-thirds power. There is a variation in the mean temperature of the superheated steam corresponding to the three velocities given in Table 4. The mean temperatures of the steam corresponding to each velocity are: 547 deg. fahr. for velocity 68.3 ft. per sec., 578.8 deg. fahr. for velocity 98.5 ft. per sec. and 458.2 deg. fahr. for velocity 29.5 to 32.8 ft. per sec. If we assume that α_1 varies as the square root of the absolute temperature and reduce the mean value of α_1 corresponding to each steam velocity to the same temperature basis, the resulting curve of α_1 as a function of the steam velocity becomes even more unexplainable, and shows first an increase and then a decrease in α_1 as the velocity is increased.

Table 5 is computed for a value of $\alpha_1 = 32$ and not for $\alpha_1 = 150$ as stated at the head of the table.

Inasmuch as Par. 17 indicates that the heat loss from superheated steam varies considerably with the velocity of the steam flow, it is believed that the steam velocities to which the data in Tables 6, 7, and 8 apply should be stated.

The comparison in Table 9 is based on the assumption that all the condensation from the saturated steam lines is wasted. It cannot be admitted that this is the general practice in power-plant or marine work. In such cases it is almost the invariable practice to return the condensation to the boilers. Any other practice would be most wasteful and would result in a poor showing for the heat cycle of the plant.

It would be interesting to know on what basis the radiation losses for superheated steam in Tables 9 and 11 were figured where the steam velocities are shown as 7000, 8000, and 10,000 ft. per min. as the highest steam velocity on which data are available in Table 4 is but 5910 ft. per min.

Fig. 1 shows that the heat loss per sq. ft. of pipe surface is greater for the 6-in. than for the 3-in. pipe. This is contradictory to the later findings of Heilman who experimented in 1922 at the Mellon Institute with 1-, 3-, and 10-in. pipes. He found the heat loss per sq. ft. of pipe surface to decrease as the pipe size increases.

Exception must be taken to the author's conclusion in Par. 22 that the heat loss per square foot is little influenced by the diameter of the pipe. Careful tests made by Heilman and reported in the paper previously referred to show a variation in heat loss per square

foot between the 1- and 10-in. bare pipes as high as 16 per cent at 100 deg. fahr. temperature difference.

Referring now to the author's summary in Par. 52, no data are given to support the heat transmission coefficient between steam and pipe wall of 400 for saturated steam referred to in sub-paragraph (*e*). The values of this coefficient found in the experiments of Clement and Garland [1] at the University of Illinois range from 1649 to 2740 and the value found by McAdams and Frost [2] at the Massachusetts Institute of Technology in 1922 was 2400.

The author's conclusion in sub-paragraph (*g*) that the coefficient of heat transmission from steam to air remains practically constant up to a temperature of 350 deg. fahr. is disproved by the work of McMillan, Bagley, and Heilman, previously mentioned, all of whom show a very large increase in the coefficient between 100 deg. and 350 deg. fahr. in the case of bare pipe. The increase in the coefficient from 100 deg. to 350 deg. temperature difference is 60 per cent according to Heilman.

Taking it all in all, there is quite a little of this data of Eberle's that is not in accord with the findings of more recent investigators. The many great benefits which result from the use of superheated steam are well known and gladly admitted, and this discussion is not intended to give any opposite impression.

Geo. A. Orrok. I think Mr. Broido is to be congratulated on his paper. He has once more emphasized that if the outside surface is kept low, and arranged so that it radiates but little, the pipe-line losses will be small.

I have had an opportunity of observing two or three heating pipe lines in which there was a covering of an inch of magnesia, over which there was asbestos board, then an air space of about an inch, then a tin shield, planished on the inside, and finally a wooden protecting pipe $1\frac{1}{2}$ in. thick. This pipe was buried in the ground about two feet below the surface and carried steam at 100 lb., and in the winter the snow did not melt above the pipe; the only case that I have known of in my experience where it was impossible to tell the location of the heating lines by the way the snow melted. In most cases the grass is green the year around over the heating pipes.

Mr. Broido has covered the velocity of steam in the pipes up to 10,000 ft. per min. This is very ordinary velocity for power

[1] Univ. of Ill. Exp. Station, Bull. 40. [2] Bull. M. I. T. vol. 57, no. 35.

plants, and in most cases the velocities are greater. It should always be borne in mind that losses are proportional to the surface and not to the amount of steam in the pipe.

J. A. Barnes. After reading the paper, the writer made comparisons slightly different from those Mr. Broido has shown and has prepared curves to illustrate them. Fig. 6 shows curves of the radiation losses per sq. ft. in B.t.u. per hr. per lb. of steam, for gage pressures of 100 and 150 lb. The upper line is the saturated steam curve, the lower one for 100 deg. superheat, and the middle for 150 deg. A curve for 200 deg. probably would fall between the

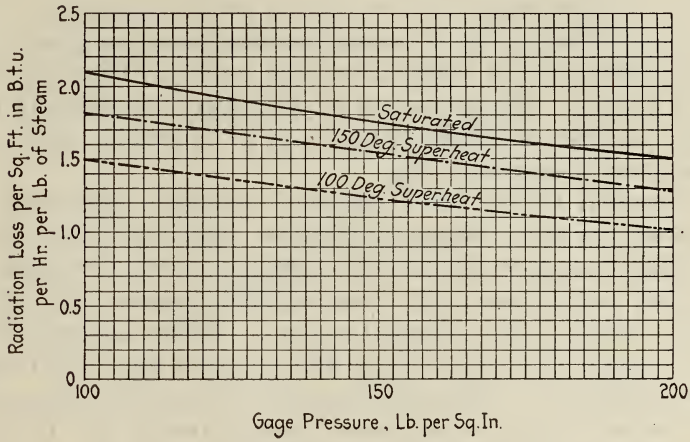

Fig. 6 Radiation Losses per Pound of Steam for Various Gage Pressures

latter and the saturated curve. It shows very clearly that they follow a given law and check closely with what Mr. Broido has reported.

With regard to the practical side, reference may be made to a power plant at Conneaut, Ohio, that originally operated with saturated steam through a pipe line between 400 and 600 feet long. At this distance it was impossible to operate the engine at anything like its full capacity. Superheaters were applied and a degree of superheat of about 100 at the engine was obtained, and the capacity was materially increased in the operation of the engine.

At the Bellman-Brook Co., a finishing company in Fairview, N. J., superheaters were installed which gave from 100 to 150 deg. superheat and the output of the plant was increased approximately

35 per cent, due entirely to the elimination of radiation losses, while practically the same thing was done in the Easton Finishing Company.

Curves of Figs. 1 and 2 of the paper seem to be contradictory. These curves represent the results of a number of tests or readings, and in plotting a curve, it is not possible to extend it through every point that is plotted. Usually such a curve is drawn to represent an average or a mean, and for that reason I assume that these curves are of this nature, and what apparently is a contradiction is merely due to the fact that they represent a series of averages.

W. H. Armacost. I have plotted some comparative curves which are shown in Fig. 7, taking the data from four papers on a similar subject, all of which have been presented before meetings of this Society. The papers are by McMillan,[1] presented in 1915, Bagley,[2] in 1918, Heilman,[3] in 1922, and B. N. Broido, which is the paper under discussion.

Curves I, II, III, and IV of Fig. 7 represent respectively the data from the above papers on radiation from bare pipes. It will be noticed that they all have the same general characteristics and that they also represent the results from numerous other investigations; that is, with saturated steam or electric-resistance-heated pipes. Curve V is plotted from data obtained from Mr. Broido's paper, Table No. 7. The curve represents the radiation from a bare pipe heated by superheated steam passing through it. It will be noted that the curve shows less radiation and also that the curve does not rise so rapidly with an increase of temperature difference as the other curves. With a temperature difference of 500 deg. fahr., a 25 per cent difference between curves IV and V will be noticed, and so on, both curves representing data obtained from the same pipe but heated with saturated and superheated steam, respectively. Consequently the curve plotted with data obtained by using superheated steam bears out Mr. Broido's statement that there is less radiation by using superheated steam in pipe lines than with saturated steam, due to the fact that saturated steam, in transferring heat to the pipe wall, partly condenses or gives up some of its latent heat, which heat is not so closely combined with the steam as the sensible heat of a liquid or the heat of superheated steam.

Comparing curves I, II, and III with curve V, it will be noted

[1] Trans., A.S.M.E., vol. 37, p. 968. [2] Idem., vol. 40, p. 667.
[3] This volume of Trans., paper no. 1848.

that the variation increases considerably with the temperature difference. The former curves were derived from data obtained by using electric-resistance-heated pipes and the latter curves by superheated steam passing through the pipe, and as a temperature difference over 400 with saturated steam is not common practice, it would not be correct to use curves I, II, and III for temperature differences above that for superheated steam, but rather curve V,

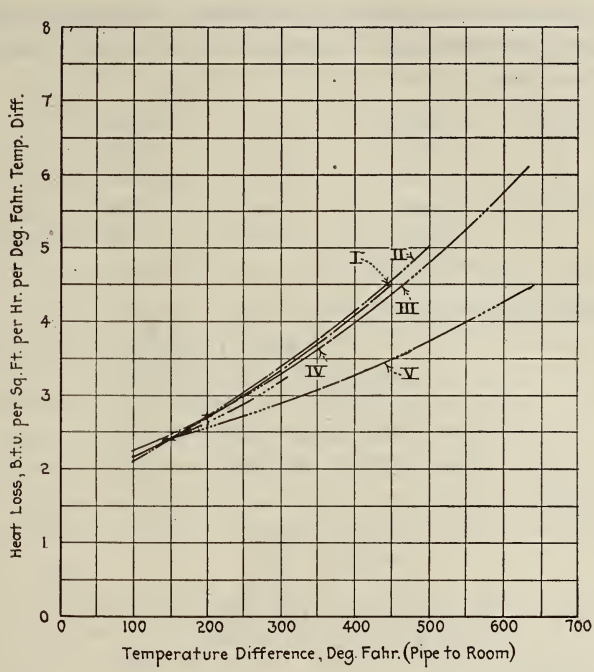

Fig. 7 Comparative Heat Losses, McMillan, Bagley, Heilman and Broido

which shows less radiation at the same temperature difference, this being brought out very clearly in Mr. Broido's paper.

C. W. Gordon. I have two questions to ask. In Par. 52*g* of the summary, the author states that the coefficient of heat transmission from saturated or superheated steam is practically constant up to 350 deg. fahr. and tends to increase at higher temperatures. In Par. 12 he states that the Stefan-Boltzmann formula may be used for calculating this coefficient. No temperature limitations are put on the use of this formula and hence further explanations appear to be in order.

In Par. 34 the author states that existing pressure drop formulas give results which are too high for straight lines carrying high-pressure superheated steam. In Par. 35, he states that the calculated pressure drop through fittings is lower than the actual test results for high-pressure superheated steam. An explanation of this would be interesting.

An attempt to check the author's proposed use of pressure-drop formulas was attempted with the following interesting results.

Applying pressure-drop formulas in the manner which the author has outlined

	Pipe Size, in.	Velocity, ft. per min.	Steam	Pressure, lb. gage	
(*a*)	4	4000	sat.	150	
(*b*)	5	4000	sat.	150	
(*c*)	5	4000	sup.	150	(150 deg. fahr. sup.)

Length = 100 ft.

Applying Babcock formula to (*a*)

$$P = 0.0001321 \frac{W^2L\left(1 + \frac{3.6}{d}\right)}{yd^5}$$

$$W = 128.6 \text{ lb. per min.}$$

$$P = 1.114 \text{ lb. per sq. in.}$$

Applying Martin formula to (*b*)

$$P = 0.0003133 \frac{W^2L}{yd^5}$$

$$W = 201.8 \text{ lb. per min.}$$

$$P = 1.12 \text{ lb. per sq. in.}$$

Applying Fritzsche formula to (*c*)

$$\frac{T}{p^w} = \frac{516 - 460}{165 \times 66.6} = 0.089$$

$$c = 1.69 \times 10^{-6}$$

$$P = \frac{1.07cw^2L}{vd}$$

$$P = \frac{1.07 \times 1.69 \times 10^{-6} \times 66.6^2 \times 100}{3.43 \times .42}$$

$$= 0.556 \text{ lb. per sq. in.}$$

A glance at the results would indicate that the author is justified in using much higher velocity for superheated steam in Table 9, with a resultant decrease in radiation loss.

J. H. Lawrence. A short time ago we designed a large central station — probably the largest in the country — Hell Gate Station, and we compared the actual losses in the steam line with the losses estimated by various formulas and while there should have been considerable drop, still when we came to measure it, it was so small that we thought there was something wrong. The loss was about one-half of the loss given by any formula in handbooks. We used no long radius bends. Every bend was made by using a fitting of some kind. Nevertheless the loss in the steam line was so small that it was almost negligible. The most serious loss was in the part where we had to provide a loop to take care of the expansion ahead of the throttle valve.

This is not the only case that has come to our attention. We have looked into this question at several stations and the loss did not check up with any of the known formulas.

We do not believe in low steam velocity. This paper mentions 10,000 ft., but this we consider too low for most steam velocity in a large station. We use a maximum of 15,000 ft., except in certain cases where the lines would only be used in case of emergency. Here we do not hesitate to go to 20,000 ft. per min. In most handbooks there are certain rules about allowable velocity, something of the order of 6000 to 8000 ft. for saturated steam and 10,000 ft. per min. for superheated steam. Such information is very misleading, as the maximum velocity should increase with the size of the pipe used.

We have found that in the case of small sizes, the velocity should be very low; for the 2-in. pipe in the neighborhood of 2000 ft. per. min.; for 6-in. pipe the proper velocity is about 6000 ft.; and so on up to 15,000 ft. velocity which we use as the ordinary maximum regardless of size of pipe.

The Author. It is very true, as has been stated by Mr. Barrett, that the analysis of radiation losses in pipe lines in this paper was not based on the experiments of McMillan, Bagley, and Heilman, all of which were made with electrically heated pipes, and the object of which was mainly to determine the efficiency of various types of pipe covering. This paper was not intended for this purpose. Its object, particularly the first part dealing with radiation losses, was to show, first, that there is a difference between saturated and superheated steam as far as imparting of heat to a pipe wall is concerned; and second, that in comparing the radiation losses of satu-

rated and superheated steam, not the direct heat in B.t.u. is to be considered, but the total heat losses in percentage of the heat conveyed in the steam through the pipe. As the Munich tests are believed to be the only extensive ones conducted with saturated and superheated steam flowing in the test pipe, only these tests could be considered and analyzed in the paper. In order to make a fair comparison, and to enable the reader to follow the analysis, the main data of the above tests were given in the paper.

Mr. Barrett looked at the paper only from the standpoint of radiation losses. I shall endeavor, however, to refer to some of the points in Mr. Barrett's discussion, as they might be of general interest.

It could not be expected that the values of α given in Table 4 would give a smooth curve, as α depends not only upon the velocity of the steam, but also upon the temperature differences. The values of α given in the table are for two velocities with bare pipe, and one with covered pipe. The temperature difference, however, with the covered pipe was considerably less than that with bare pipe.

Mr. Barrett objects to the heat transmission coefficient between the steam and pipe wall given as 400 in Par. 52, because other investigators have found it to be considerably higher, or up to 2700. From Table 1 of this paper it is seen that at a temperature difference between the steam and the outside air of 250 deg., the heat loss per square foot of surface per hour in B.t.u. is 750. With a heat transmission coefficient of 400, the temperature of the wall must be $750 \div 400 = 1.875$, or less than 2 deg. below the steam temperature. At a wall temperature equal to the steam temperature, α would be infinitely large. As it is very difficult to measure accurately the temperature of the steam and the pipe wall, and a difference of 2 deg. is likely to occur, the value of α cannot be definitely fixed. 400 is mentioned in this paper only as a comparison with the value of superheated steam, which is only 32 at a velocity of about 80 ft. per second.

As shown on the curves plotted by Mr. Armacost while discussing this paper, the difference between the data of the tests analyzed in this paper and those of Messrs. McMillan, Heilman, and others, is very slight.

Mr. McMillan wonders why the difference between the heat transmission of saturated and superheated steam was gone into in such detail. As mentioned in Par. 18, this subject is of considerable interest as far as the use of superheated steam for heating and

drying purposes is concerned. So far as the author is aware, the influence of the velocity of superheated steam on the heat transmission, particularly in connection with the question of whether the pipe is wet or dry inside, has never before been discussed, and is believed to be of considerable interest to engineers.

Mr. McMillan's statement that calculating steam line sizes on velocity is antiquated, and that the preferred method of calculating steam lines is based on allowable pressure drop, is not quite understandable, as the pressure drop is dependent on the velocity of steam, so that the results must be the same. In Par. 30 is mentioned why it is more advisable or possible to use higher velocities with superheated steam. The velocity taken in Table 9 is in accordance with general engineering practice. That the pressure drop with these velocities is not excessive, is shown by the fact that it is only slightly greater than that for saturated steam, and with the exception of one case does not exceed 1 lb. throughout the range of the table.

In answer to Mr. Orrok's question as to how the wall temperatures of the pipe were measured, thermocouples were placed just on the outside of the pipe, and held by Z-shaped bars, which were formed to fit snugly to the pipe, and being in metallic contact with the wall, obtained the same temperature as the pipe. The thermocouple was pressed to the pipe by these bars, and in this way the element was practically imbedded in metal having the temperature of the wall. With this arrangement, if carefully carried out, very reliable readings can be obtained, while if the element is only laid on the pipe, considerable errors may result.

When Mr. Lawrence, in comparing the pressure drop obtained at Hell Gate with that given in formulas, found that it was only one-half of that obtained by calculations, he probably figured with the old Babcock or Guttermuth formulas. If they were compared with figures obtained with the Fritzsche formula, as mentioned in the paper, they would probably check more closely. As a matter of fact, on several occasions I had opportunity to compare the pressure drop obtained by the formulas of Babcock and Guttermuth with that of Fritzsche; for large pipes particularly, the results obtained with the Fritzsche formula were only about one-half of the others, which rather checks with the findings at the Hell Gate Station.

The fact that the Stefan-Boltzmann formula may be used for calculating of radiation loss from pipe lines is not contradicted by

the statement that the coefficient of heat transmission from saturated or superheated steam is practically constant up to a temperature difference of 350 deg., and that it tends to increase at higher temperatures, as was questioned by Mr. Gordon. In accordance with the Stefan-Boltzmann formula the heat transmission varies with the fourth power of the temperature difference, so that it varies little at lower differences, and increases rapidly at the highest, and therefore, confirms rather than contradicts the statement.

With reference to the second question of Mr. Gordon as to why the friction in pipe lines at high pressures is less, while in fittings it is higher than that derived by calculating according to standard formulas for moderate steam pressures; this is probably due to the fact that in fittings the steam changes its direction, and a part of its kinetic energy is lost, and this is more pronounced at higher pressures than at the lower ones.

No. 1876

TEN YEARS' PROGRESS IN MANAGEMENT

By L. P. Alford, New York, N. Y.
Member of the Society

TEN YEARS have passed since the Committee report on The Present State of the Art of Industrial Management was presented to The American Society of Mechanical Engineers. The request is now made for a review of the progress of management during the intervening decade. Unfortunately for the purpose of such a study, eight of these ten years were abnormal, many of the management changes and innovations introduced were of a temporary nature or were mere expedients, and it is difficult to separate them from other and more permanent developments.

2 The only satisfactory way to treat the review is to base it upon the report of 1912, which was well received and in large measure approved. This course has therefore been adopted.

3 At the outset we should recall and pay generous tribute to three of our late great leaders who aided in preparing that report and took part in its discussion: Frederick W. Taylor, the pioneer in management; Henry L. Gantt, who humanized the movement; James M. Dodge, the earnest, constructive supporter. During our ten-year period these men of vision and power completed their life work.

4 To obtain information on the worth-while changes which have taken place, letters were written to management and industrial engineers, to executives of plants in various lines of industry, and to educators familiar with industrial developments. Many interviews were held with men having industrial and managerial

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responsibilities. The response to these requests has been most generous. The author is deeply indebted for the information received and expresses his sincere gratitude to all who have given aid.

5 The report of 1912 declared the new element in management to be: "The mental attitude that consciously applies the transference of skill to all the activities of industry." It also quoted[1] and endorsed three regulative principles:

a The systematic use of experience
b The economic control of effort
c The promotion of personal effectiveness.

6 New interpretations and expanded meanings have been given to these principles, but they have in nowise been weakened or superseded. One correspondent writes: "Note, for example, the nearly universal acceptance of the principles . . . ."

7 In answer to the question, "What steps have been made in the progress of management since 1912?" a wide range of opinion was expressed as shown by the following sixteen quotations from correspondents' letters. The first gives a particularly well-balanced judgment of the situation.

> It seems to me that management has very definitely progressed in the last ten years along certain main lines.
>
> In the first place, good management is more insistent today on knowledge as a basis of judgment, rather than the old judgment based on personal observation. Management is more and more demanding costs, a knowledge of inventories, monthly profit and loss statements, statistics, and records of all kinds as pictures of events on which to base judgment.
>
> In the second place, management is now undergoing a definite metamorphosis in the matter of industrial relations, and managers are waking up to the fact, as a practical element in their business, that they owe more to their employees than mere wages, and that whistle blow and hustle are not all there is to factory operation.
>
> It is this belief and the spirit developing, rather than the volume of the action up to date, which is a matter of very definite progress in the past ten years of management.

8 Ten opinions, three to the effect that management has retrogressed or made little or limited progress, and seven stating the belief that progress has been made and mentioning certain details of improvement, are grouped to present a contrasting though in the main favorable picture.

[1] *American Machinist,* vol. 36, p. 857, The Principles of Management, by Church and Alford.

Management (the directing group) has retrogressed in its acceptance of the principles of management, while labor has materially progressed toward a broader acceptance of these principles.

I believe that very little progress has been made in the adoption of scientific management principles in industries outside of metal working, with a few notable exceptions.

The main advance, and that lamentably slow, has been in putting into practice knowledge already available previous to 1912.

During the past ten years we have passed through the period of first glamour, then the reaction of a loss of confidence, and have finally evolved into the general recognition of the legitimate place of a new branch of engineering art and science — management engineering.

The important steps in progress in management during the past ten years have been from unintelligent rule-of-thumb management through scientific management to intelligent management. The latter has advanced steadily during the decade.

The greatest progressive step has been toward standardization of appliances and methods.

The most definite progress made during the past ten years is the universal acceptance of the merits of specialized production and standardization of design. These two steps have opened the way to a third simplification of method.

The reaction from destruction and waste incident to warfare and reconstruction has been toward the elimination of waste in industry as a management function. Waste in all forms has been more closely observed than hitherto, especially during the past two years. This effort to do away with waste has led to the fixing of budgets and the determination of cost standards.

Important steps in the progress of management since 1912 are:

a Greater use of facts in the establishment of the standards by which business is conducted;

b Broader recognition of the principle that industry exists for service to humanity;

c Greater appreciation of the importance of regularization or control in the successful conduct of our industries;

d Wider understanding of the economic value and importance of the management engineer in the operation of business.

There has been a great increase in the use of specifications not only to govern purchased materials but also to attain uniformity of process, quality and cost, and thus to insure reliability of product. Many plants now have well-equipped laboratories staffed by scientific men and some regularly employ consulting scientists. In the larger corporations research laboratories are not uncommon. Few of these departments are over ten years old and they evidence a rapidly growing appreciation of pure science as a tool of management.

The need of early and reliable figures as a mechanism of management has caused many companies to prepare monthly a complete statement of their business and earnings. A constantly increasing number of companies are publishing annually a detailed statement of their financial condition and many are publishing such statements quarterly. This voluntary publicity indicates a sincerity and frankness rare in management of an earlier decade.

9 The final quotations in regard to progress, five in number, discriminate between management form and substance. Progress is indicated in both of these aspects.

The biggest and most lasting accomplishment in the inculcation of management principles is that, like religious teaching, whose significance has been forgotten during years of prosperity, they again, in the years of depression following the war, developed a new significance in the minds of the thoughtful. A principle is not established in the actual social inheritance of the race — as a step forward — until man has applied it to himself and seen whether its application makes him a better man in his social relations. So management principles are being used as yardsticks to measure individual industrial development. This means that these principles are becoming a subconscious part of the mental equipment of industry, and not only is this real progress, it is fundamental.

The development and use of the Gantt chart is the most important step of progress, because it calls attention to the movement of facts, to the necessity of basing decisions on facts rather than on opinions, and because it helps managers to foresee future happenings.

A second important step is the change in the method of installation from the old type which organized from the top down to the new type which builds from the bottom up.

A third important step is the development of the theory that the cost of an article includes only those expenses actually incurred in the production of the article, and that the expenses of maintaining one machine in idleness cannot be charged into the cost of the output of another machine. Along with this theory came the development of a method of arriving at costs of idleness and work.

Probably the greatest progress consists in a better understanding of the problems of management with a particular acceptance of the facts to which Taylor called attention, that management is an art which may be practiced advantageously through the application of certain principles and the scientific method. I do not think, as yet, that the great majority of men at the head of industries have anything like an adequate understanding of scientific management, nor that they are able to distinguish between form and substance in this respect. They have, however, apparently emerged from the attitude of opposition and mistrust of so-called scientific management, but are satisfied with a superficial application of the principles of management.

The important steps of progress made in management since 1912, I would say, are as follows. The order in which they are named is not significant:

a A greater appreciation of the human factor in industry.

b The growing recognition that employees should have a voice in the management as relating to those questions that directly affect them.

c The recognition of the strategic position of shop foremen and the necessity of more carefully selecting and training them.

d The increased recognition of the value of fundamental principles.

e The recognition of, and in a large degree the adoption of, standard systems of cost accounting from the point of view of timeliness, as a barometer rather than history, as an instrument of production rather than a matter of finance.

f A great development in mechanical equipment, combined with improved plant layout and building plants to fit manufacturing process.

g A marked advance in sales policies.

h A marked advance in substituting the trained, competent engineer for the old " cut-and-try " type of executive.

Using figures, which after all are most impressive, but basing those figures purely on my impressions, I would say that since 1912 industry has progressed in management by some 30 to 40 per cent in the appreciation of the fact that there is a management problem aside from the old concept, which was that the owner had simply to censor the things that happened within his jurisdiction. I should say that there had been a 20 to 25 per cent endeavor to install the mechanisms of management, considering in this figure the generally known stores systems, operation studies, wage-incentive plans, etc. In some cases, as for instance in stores control, the percentage might run a great deal higher, but I am refraining from increasing my estimate, for it is my belief that these mechanisms that we have installed are, for the most part, of a makeshift character, and that in industry as a whole and considering only the larger companies, I doubt if more than five or six per cent are possessed of mechanisms at all acceptable in the final scheme of what management should do and possess.

As to the real concept existing today of what management is, and what conditions must be considered, influenced and coördinated to bring about the situation which should exist, I doubt if more than one-fifth or one-fourth of one per cent of the companies in this country possess a knowledge or even appreciation of what is real management.

10 Combining and weighing these carefully prepared statements and adding to them certain well-recognized facts, there emerges a group of factors of varying importance which mark the progress of management during the past decade. These naturally arrange into three groups, of which the first concerns changes in mental attitude.

a The ancient controversy as to whether management is a science or an art has subsided, with increased recognition of the scientific basis of management.

b The attitude of opposition and mistrust toward management and the passionate antagonism to the installation of management methods have in general disappeared.

c Among those responsible for the carrying on of industry there has grown an appreciation of the existence of problems of management. (The appointment of Herbert Hoover as Secretary of Commerce and General Dawes as Director of the Budget reflect an appreciation by the Government of the need for management in our national affairs.)

d Acceptance of the principles of management has broadened among engineers, executives in industry, and educators.

11 The second group of factors of progress concerns the application of management methods.

e The engineering or scientific method has extended in industrial cost accounting. Among the developments are uniform cost-accounting systems (64 manufacturers' associations which have adopted such systems), the theory and method of determining and applying standard costs, the methods of determining idleness losses, the forecasting of sales leading to long-term production schedules, and the budgeting of future expenditures.

f Appreciation of the possibilities and advantages of standardization, simplification, and elimination of waste has spread rapidly during the past two years.

g The demand for knowledge, facts, as a basis for judgment has grown insistent in all good management. This has led, among other developments, to a widespread use of specifications and graphics as a means of recording and communicating management knowledge. (The first modern book on graphics in the English language was published as recently as 1910. The Gantt-type control chart has been developed into its present form since 1917.)

h Management methods have been applied or installed in practically every manufacturing industry, in distributing concerns and in institutions. (The report of 1912 listed 52 industries in which some form of management had been installed. A similar list prepared in 1922 would group all the branches of American industry.)

12 The third and final group of these factors concerns especially significant developments, which after being stated are subject to explanation and comment.

i Management activities have broadened far beyond the installation of those mechanisms which are usually associated with the Taylor System, and which were emphasized in the report of 1912. (Appendix No. 1 lists 77 items of management work arranged under four headings: General, Labor, Material, Equipment.)

j Some eight or ten of the leading American engineering schools have established courses in management since 1912.

k Appreciation of the importance of the human factor in industry and attempts at its study from a fact basis have been the most striking management development.

l Management engineers have declared that the service motive must prevail in industry and that all questions concerning human relationships must be considered in a spirit devoid of arbitrariness or autocratic feeling.

MANAGEMENT MECHANISMS

13 To secure information as to the use of management mechanisms, the question was asked, "What (if any) mechanisms of management do you consider as generally accepted (*a*) in principles, (*b*) in practice?" From correspondents' replies the following twelve quotations have been selected.

I do not believe that any of the mechanisms of management are generally accepted in principle or in practice.

I know also that even where some of these things (mechanisms of management) have been established and we hear about them and might conclude that the firm using one or more of them is quite advanced, it often is not at all so. The feature described is only an unrelated "stunt," not supported by a complete coördinated system of administration and usually begins to go to pieces not long after it is installed.

There is at the present time a retrograde movement in regard to the building up of stores and making operation studies. However, as I see it, this is merely a temporary depression in the curve, and I believe that the general tendency of this curve is upward with a very slow ascending grade.

Incentive wage-payment plans have had a temporary setback due to labor conditions caused by the war and to the reluctance of managers in general to consider such plans in any other light but of profit to the company. I do not think that the main service which the incentive plan can give — namely, that of stabilizing relations between employers and employees — has been given sufficient attention by the management.

I believe that such mechanisms as balance of stores, routing, operation studies, incentive wage plans, personnel work, etc., are generally accepted in principle, but that efforts to install them frequently (perhaps most of the time) miscarry, and either accomplish little or no good. This is often due to a failure to see to it that details connected with the mechanisms are fully understood and looked after.

In a general way, the mechanisms of management are widely accepted now in principle and much less widely in practice.

Undoubtedly, good storeskeeping is becoming very generally accepted. We know that unless we keep accurate records of the materials used we cannot get the most satisfactory results. I think storeskeeping is accepted both in principle and practice as well as the intelligent study of operations.

I believe that balance of stores is accepted in principle and in practice, that is, in so far as a written record of quantities in stores is kept in the office rather than in a storesroom, and that a minimum or order point is predetermined and an order placed when it is reached. It is generally accepted in principle, though not in fact, that an incentive wage-payment plan is desirable and effective. It is accepted in principle that facts are shown on charts better than in tables of figures.

In principle, undoubtedly, all of the main mechanisms of management have been thoroughly established.

In a great measure all of the mechanisms of management as developed by Taylor and his immediate associates have been generally accepted in principle. But while they are being widely applied, my impression is that in the great majority of the cases the application is half-baked in character, and the results, while they may be satisfactory to the companies concerned, are far from being as satisfactory as they should be, either to the management or to the employees. My experience indicates that in most such cases an application such as Taylor would have approved will almost invariably result in increased production ranging from 30 to 100 per cent or more, depending on the nature of the business.

The following management mechanisms have been accepted in varying degrees:

Stores Control. In principle and practice very generally.

Operation Standardization. (*a*) In technical aspects, generally in principle, fairly so in practice. (*b*) In personal aspects, fairly accepted in principle, to a limited extent in practice. [By (*a*) I mean speeds

feeds, equipment, tools, etc.; by (*b*) motion and time studies of human elements.]

Wage-Payment Incentives. Generally in principle and in practice so far as direct labor is concerned. But little application has been made to indirect labor.

Cost Accounting. Generally accepted both in principle and practice.

Selection and Training of Employees. Fairly well accepted as to principle, but little in practice.

Purchasing Control. Generally as to both principle and practice.

Scheduling and Planning. Fairly well accepted in principle. Limited in practice in some industries, well established in others.

I find mechanisms being accepted one by one without a full realization of the part they are to play in the scheme as a whole. That is to say, I will find a company suddenly appreciative of the value of operation studies. It will thereupon proceed to organize to make operation studies, and for the time being in its new enthusiasm it pursues what threatens to become a hobby rather than a part of its business. This pursuit at times leads into the installation of other mechanisms. It begins to recognize, from the operation studies, that a balance of stores is essential, and that a wage incentive is desirable. I find, however, that this progress is accidental rather than planned.

Mechanisms of management such as are discussed in the 1912 Report are generally accepted in principle, but poorly carried out in practice in the majority of establishments. On the other hand, a few, representing the best organizations, have developed these things to a degree which serve as valuable guides.

Planning and control are used more and more extensively in plant operation. The tendency of the majority, however, is to try to gain the benefits of more intensive control through partial makeshifts which record past accomplishments instead of actually planning the work. The importance of control, in fact, in increasing production through elimination of idle time, men, and machinery, is not yet recognized except in a few markedly successful establishments. The developments along these lines are being undertaken frequently through inexperienced, low-grade men, who adopt mechanisms as such, instead of developing existing methods on fundamental principles.

Balance of stores is accepted almost universally in principle and widely used in practice. Accountants have been quick to recognize its advantages, and have made it an essential part of their accounting mechanism. On the other hand, two of the most vital features for assisting in the control of production, the column of "stores apportioned" and the entering of "minimum" quantities of each item permissible, are apt to be omitted.

The development of time study and job analysis, while wide-spread, has been unsatisfactory; piece rates are more and more universal, but their determination is still largely on a basis of past performance, aided by time studies which simply record these performances in more detail instead of analyzing the operations and determining the methods and units which will

give most satisfactory results. There is still lack of appreciation of the fact that the chief aims of time study and job analysis must be:

a To resolve the operations into such units that they can be recombined to provide for all variables;

b To take advantage of this unit study to eliminate unnecessary operations, substitute improved methods, and remove defects in equipment and in control;

c To enable the workman to earn more money often with less effort; and,

d To indicate means for improvement in quality and practicable methods for making the improved quality routine.

14 To these statements of the acceptance of management mechanisms it is possible to add a few quantitative facts. It will

TABLE 1

Name of Industry: *Metal Trades.* No. of Questionnaires: 16

NOTE: The wide diversity in kinds of work, varying from machine-shop to shipyard, must be borne in mind, also a wide variation in size of plants.

Mechanisms of Management	None	Inadequate	Good	Questionnaire Reference
1 Selection and Placement	0	6	10	K7
2 Incentive Wage Plan	2	6	8	K13
3 Planning Centralized	3	8	5	U4
(*a*) Routing, order of work	5	6	5	U4. 9
(*b*) Schedules, machine assignments	4	5	7	U5. 6. h
4 Time Study	6	2	8	U4. ef
5 Cost Control	4	3	9	U7
6 Idle-Time Analysis:				
(*a*) Men	13	0	3	U8
(*b*) Machines	9	1	6	U3
7 Purchase Control	3	4	9	U11
8 Balance of Stores	2	4	10	U11, 12

be recalled that the field reports of six industries, given in the Report of the Committee on the Elimination of Waste in Industry of the Federated American Engineering Societies, were based on an extensive questionnaire. The replies in four of these industries — metal trades, boot and shoe manufacturing, men's ready-to-wear clothing manufacturing, and printing — have been studied to show the use of mechanisms of management. The facts brought forth are presented in Tables 1 to 6, inclusive. Table 1 records the results for 16 metal-trades plants where the entire questionnaire was used; Table 2 is from 12 metal-trades plants where a condensed questionnaire was used; Table 3 is from 8 boot and shoe shops;

TABLE 2

Name of Industry: *Metal Trades.* No. of Questionnaires: 12

NOTE: This is additional information from 12 plants assayed more briefly than the 16 of Table 1. A condensed questionnaire was used.

Mechanisms of Management	None	Inadequate	Good	Questionnaire Reference
1 Selection and Placement	0	4	8	No. 10
2 Incentive Wage Plan	1	3	8	No. 13
3 Planning Centralized	1	2	9	No. 19
(*a*) Routing, order of work	2	2	8	No. 19
(*b*) Schedules, machine assignments	1	3	8	No. 19
4 Time Study	2	1	9	No. 19
5 Cost Control	2	6	4	No. 18
6 Idle-Time Analysis:				No. 22
(*a*) Men	8	0	4	
(*b*) Machines	6	0	6	
7 Purchase Control [1]	1	1	10	No. 4
8 Balance of Stores	1	1	10	No. 4

[1] Purchase Control *as such* was not covered in the condensed questionnaire, but this estimate was based on the general stock and stores systems; whether there were maxima and minima in stores, and a control of raw material all through.

TABLE 3

Name of Industry: *Boot and Shoe Mfg.* No. of Questionnaires: 8

Mechanisms of Management	None	Inadequate	Good	Questionnaire Reference
1 Selection and Placement [1]	0	6	2	K7
2 Incentive Wage Plan [2]	0	0	8	K13
3 Planning Centralized	4	0	4	U4
(*a*) Routing, order of work	4	1	3	U4, g
(*b*) Schedules, machine assignments	3	3	2	U5, 6, h
4 Time Study	5	2	1	U4, 3, f
5 Cost Control	1	3	4	U7
6 Idle-Time Analysis:				
(*a*) Men	7	1	0	U8
(*b*) Machines	7	1	0	U3
7 Purchase Control	1	4	3	U11
8 Balance of Stores	1	1	6	U11, 12

[1] Not important, as skilled operatives are always available.
[2] Union control very strong in this industry.

TABLE 4

Name of Industry: *Men's Ready-to-Wear Clothing Mfg.* No. of Questionnaires: 9

Mechanisms of Management	None	Inadequate	Good	Questionnaire Reference
1 Selection and Placement [1]	0	4	5	K7
2 Incentive Wage Plan [2]	4	2	3	K13
3 Planning Centralized	4	2	3	U4
(*a*) Routing, order of work	4	2	3	U4, 9
(*b*) Schedules, machine assignments	3	2	4	U5, 6. h K13
4 Time Study	3	3	3	U4, e, f
5 Cost Control	2	5	2	
6 Idle-Time Analysis:				
(*a*) Men	6	3	0	U8
(*b*) Machines	8	1	0	U3
7 Purchase Control	3	1	5	U11
8 Balance of Stores	3	1	5	U11, 12

[1] Not important, as skilled operatives are always available.
[2] Union control very strong in this industry.

TABLE 5

Name of Industry: *Printing.* No. of Questionnaires: 6

Mechanisms of Management	None	Inadequate	Good	Questionnaire Reference
1 Selection and Placement [1]	0	5	1	K7
2 Incentive Wage Plan [2]	1	1	4	K13
3 Planning Centralized	3	2	1	U4
(*a*) Routing, order of work	5	0	1	U4g
(*b*) Schedules, machine assignments	3	2	1	U5, 6, h
4 Time Study	5	0	1	U4, e, f.
5 Cost Control	3	2	1	U7
6 Idle-Time Analysis:				
(*a*) Men	5	0	1	U8
(*b*) Machines	2	0	3	U3
7 Purchase Control	3	2	1	U11
8 Balance of Stores	4	1	1	U11, 12

[1] Not important, as skilled operatives are always available.
[2] Union control very strong in this industry.

Table 4 from 9 men's ready-to-wear clothing shops; Table 5 from 6 printing establishments; Table 6 is a summary for the 51 plants studied. The questions from whose replies the facts were drawn are given in Appendix No. 2.

15 Turning to Table 6 and arranging the eight mechanisms

TABLE 6 SUMMARY FOR 51 PLANTS IN 4 INDUSTRIES

Mechanisms of Management	Boot & Shoe, 8 plants			Men's R. M. Clothing, 9 plants			Printing, 6 plants			Metal Trades, 28 plants			Totals, 51 plants		
	None	Inadequate	Good	None	Inadequate	Good	None	Inadequate	Good	None	Inadequate	Good	None	Inadequate	Good
1 Selection and Placement	0	6	2	0	4	5	0	5	1	[1]0_0	6_4	10_8	0	25	26
2 Incentive Wage Plan	0	0	8	4	2	3	1	1	4	2_1	6_3	8_8	8	12	31
3 Planning Centralized	4	0	4	4	2	3	3	2	1	3_1	8_2	5_9	15	14	22
(a) Routing, order of work	4	1	3	4	2	3	5	0	1	5_2	6_2	5_8	20	11	20
(b) Scheduling, machine assignments	3	3	2	3	2	4	3	2	1	4_1	5_3	7_8	14	15	22
4 Time Study	5	2	1	3	3	3	5	0	1	6_2	2_1	8_9	21	8	22
5 Cost Control	1	3	4	2	5	2	3	2	1	4_2	3_6	9_4	12	19	20
6 Idle-Time Analysis:															
(a) Men	7	1	0	6	3	0	5	0	1	13_8	0_0	3_4	39	4	8
(b) Machines	7	1	0	8	1	0	3	0	3	9_6	1_0	6_6	33	3	15
7 Purchase Control	1	4	3	3	1	5	3	2	1	3_1	4_1	9_{10}	11	12	28
8 Balance of Stores	1	1	6	3	1	5	4	1	1	2_1	4_1	10_{10}	11	8	32

[1] The two figures shown separately in the metal-trades columns represent totals for the 16 plants (upper figure) covered by the regular questionnaire, and the 12 plants (lower figure) which filled out only a condensed questionnaire.

in the order of the number of plants in which they are installed *in some form,* we have:

Selection and Placement
Incentive Wage Plan
Balance of Stores
Purchase Control
Cost Control
Planning (routing, scheduling)
Time Study
Idle-Time Analysis.

Rearranging in the order of the number of plants where the installation is *good,* we have:

Balance of Stores
Incentive Wage Plan
Purchase Control
Selection and Placement
Planning (routing, scheduling)
Time Study
Cost Control
Idle-Time Analysis.

16 The weight of opinion and fact brings the conclusion that certain mechanisms of management have made decided headway in acceptance both in principle and practice, and from an assay of four industries the importance of application yields two groups:

a Balance of Stores
Incentive Wage Plan
Purchase Control
Selection and Placement

b Cost Control
Idle-Time Analysis
Planning
Time Study.

17 In the installation of such mechanisms a significant change is becoming evident. In the early days of management the mechanisms concerned the physical means of production. They were originated by the executives and were ordered into the shop.

18 At a later date, as emphasized in the report of 1912, the value of methods which concerned the worker was appreciated. Training was the first to have any widespread trial. But the attitude was still the developing or forcing of a mechanism from the top downward.

19 Within the decade under review, another attitude has been adopted in a few instances. It seeks to make the foremen and even the workers consciously parties to the development of the plans before they are put into effect. It endeavors to arouse interest, to inspire to achievement, to release creative energy. Its effect is to install methods and mechanisms from the bottom upward with celerity and improvement in personnel relations.

MANAGEMENT EDUCATION OF ENGINEERING GRADE

20 Where there was probably but a single college course in management in 1912, there are now eight, in these institutions:

Columbia University
Massachusetts Institute of Technology
New York University
Pennsylvania State College
Purdue University
University of Kansas
University of Pittsburgh
Yale University — Sheffield Scientific School

In addition to this form of instruction, management subjects have been introduced in mechanical-engineering courses. Examples are the pioneer work at Cornell University — Sibley College, and at the Worcester Polytechnic Institute.

21 The growing importance of this branch of engineering education is shown by the number of men enrolled. Appendix No. 5 gives the enrollment of all students in colleges of engineering in the United States for the school year 1921–22. The total is 53,414. The number in management courses is 1123, identified as

Administration Engineering	725
Industrial Engineering	389
Industrial Management	9
Total	1123

22 The 277 students in "Commercial Engineering" courses have not been included, although they undoubtedly received some instruction in management subjects.

23 While these management courses in the beginning were based on mechanical engineering, their character seems to be changing, so that it can now be said that they are based on engineering broadly, with emphasis on fundamental subjects. There is a tendency to lessen or limit qualitative instruction in details of production. Without doubt, the character of the instruction is improving as teachers gain a wider and sounder experience in the application of management principles.

24 The significance of this new branch of engineering education is not its extent as measured in numbers of students, but in the fact that at least eight leading institutions have added it to their regular and older courses.

25 It is unfortunate that no common name has been adopted for these courses; at least four are in use.

THE HUMAN FACTOR IN INDUSTRY

26 The report of 1912 presented the human factor in industry with particular emphasis on the responsibility of managers and executives to train their workers and the same thought was prominent in the discussion. According to the comment of the Committee in its closure, one of the striking characteristics which had already gripped attention was "the presence throughout the dis-

cussion of a human spirit in keeping with the best trend of thought toward social justice," and "the development that has taken place within the last few years leading to a new appreciation of the needs and rights of employees."

27 Henry P. Kendall in his discussion of the report [1] outlined the operation of an employment department which he had initiated. The employment man interviewed applicants, selected workers by tests, placed them in positions for which they were fitted, required medical examinations, kept records of each employee, kept in touch with the foremen in regard to the deportment, skill and earning power of the employees, had charge of discipline and discharge, and gave advice, suggestions, and sympathy to the workers.

28 These disclosures in outline foreshadowed a great wave of industrial relations work which swept through American industry after the outbreak of war. The movement received its impetus from the demand for workers in a time of extreme shortage, and was influenced by emotionalism and social theory. With the return of a labor surplus in 1921 the unsound features have in the main disappeared, leaving but vestiges of the methods and devices which were initiated in such profusion.

29 The present situation as regards personnel work is appreciation that personnel problems exist, recognition that their solution is a responsibility of management, and a growing realization that job analysis, selection, placement, and training can be put on a scientific basis.

30 Associated in thought, though not necessarily a part of any employment or industrial relations plan, is the rise of works councils in American industry. Several hundred have been established during the past decade. In August, 1919, there were 225, in February, 1922, approximately, 725.[2] Their development has been in response to a desire on the part of the workers for a means of expressing their beliefs and wishes in regard to matters arising in employment, and on the part of the management for a means of communicating with their employees and gaining and holding their confidence and good will. The movement but emphasizes the fact that the development of the relationships of employer and employed is a responsibility of management.

[1] Trans. Am. Soc. M. E., volume 34, p. 1208.

[2] See Reports of the National Industrial Conference Board.

THE SERVICE MOTIVE

31 Management engineers as a group have declared that the service motive must prevail in industry, that everything planned and done must be directed to securing the worthy result of producing useful goods with a minimum expenditure of time, material, and human effort. One of the clearest statements was written by Henry L. Gantt a few weeks before his death: [1]

> We have proved in many places that the doctrine of service which has been preached in the churches as religion is not only good economics and eminently practical, but, because of the increased production of goods obtained by it, promises to lead us safely through the maze of confusion into which we seem to be headed, and to give us that industrial democracy which alone can afford a basis for industrial peace.

32 This disinterested purpose has been accepted as an ideal for the entire engineering profession, by becoming the challenging thought in the preamble to the constitution of the Federated American Engineering Societies.

33 To the factors dealing with the steps in the progress of management which have been discussed, should be added a consideration of the extension and growth of management societies which has taken place during the past ten years.

34 The earliest was the Society for the Promotion of Scientific Management, founded informally in 1910 and organized in 1912. In 1916 the name was changed to the Taylor Society, and in 1918 it was reorganized. See Appendix No. 3.

35 The first national organization to deal with personnel matters beginning with the training of workers was the National Association of Corporation Schools, founded in 1913. By 1917 its work had broadened to include all of the activities classified as human relations. In 1920 the name was changed to the National Association of Corporation Training. In May, 1922, it was merged into the National Personnel Association. See Appendices Nos. 4 and 8.

36 In May, 1917, in response to a war demand, The Society of Industrial Engineers was founded. In 1919, it was functionalized and has carried on the activities of a professional engineering society. See Appendix No. 5.

37 A second personnel society was organized during the war, May, 1918, under the name of the National Association of Employ-

[1] Organizing for Work, p. 104.

ment Managers. On March 1, 1920, the name was changed to The Industrial Relations Association of America. May of that year registered the peak of the movement, the Chicago National Convention being attended by 5000 persons. The change in business conditions affected it adversely and in December, 1921, the Board of Directors voted to disband the organization. Early in 1922 it was merged into the National Personnel Association. See Appendices Nos. 6 and 8.

38 Although The American Society of Mechanical Engineers provided the forum for the presentation of the earliest papers on management, no part of that society was particularly devoted to management matters until the formation of Professional Divisions in 1920. In July of that year, the Management Division was organized. It soon led all of the other Professional Divisions in membership and has held that position ever since. See Appendix No. 7.

39 There are, therefore, four societies concerned with management — three in its engineering or technical aspects, and one restricted to personnel matters. The combined membership, not discarding duplications known to exist, is 4041.

Management Division A. S. M. E.	1,740
Society of Industrial Engineers	1,032
The Taylor Society	769
National Personnel Association[1]	500
Total	4,041

40 This membership is growing rapidly; more than one-half has been gained during the past two years, for that period spans the founding and growth of the management division of The American Society of Mechanical Engineers.

41 Within the last two years joint activities have been originated among these and other societies with the promise of benefits to all who are concerned with management. Included are: Development of a management terminology; development of a classification for management literature; standardization of management graphics; and development of methods for the measurement of management.

[1] The National Personnel Association also has 129 company members.

MANAGEMENT RESULTS

42 The report of 1912 stated that the results of good management had been: "A reduced cost of product, greater promptness in delivery with the ability to set and meet dates of shipment; a greater output per worker per day with increased wages; and an improvement in the contentment of the workers." There was no evidence at that time that goods had been reduced in price to the consumer.

43 To a degree this evidence has now been supplied. There are examples where good management has held down prices during a period of inflation and reduced prices as soon as business conditions changed. These acts benefited the consumer. Therefore the management movement has earned its economic justification.

44 Management as developed through a generation of effort stands today as a great body of knowledge and practice to facilitate the operation of industry and the conduct of business. Through organization it determines policies, plans basically over long periods of time and fixes impersonal relationships; through preparation it plans in detail how, when, and by whom work is to be done; through direction it initiates and maintains the processes of production and distribution.

45 Here, then, is a tremendous, hitherto unknown engineering tool. What is it for? The answer is a spur to every engineer and industrial executive:

46 *Industry and business* as developed in modern civilization must continue, else infinite misery will overtake the human race. Management is the agency by which community, state, and nation shall endure.

APPENDIX NO. 1

MANAGEMENT WORK

47 An incomplete list of management work follows, arranged under four headings — General, Labor, Material and Equipment:

General

Reduce Kinds and Varieties of Product
Stabilize Production
Determine Best-Product Design
Locate Unprofitable Lines of Product
Develop New Lines of Product
Set Basic Production Rates
Establish Production Control
Organize or Reorganize Plant
Develop Organization Charts
Coördinate Sales Effort with Factory Conditions
Predetermine Selling Costs and Prices
Improve Deliveries to Customers
Prepare Graphical Statements for Executives
Provide Plans for Executive Conferences
Prepare Financial Statements
Establish Idleness Records
Standardize Office Procedure
Prepare Standard Practice Instructions
Define Engineering and Drafting-Room Procedure
Develop Control Accounts
Determine Standard Costs
Capitalize Suggestions
Establish Financial Aids for Workers
Institute Safety Campaigns
Establish Employees' Committees and Representation.

Material

Organize Material Storage, Records and Control
Prepare Bills of Material or Parts Lists
Determine Materials Specifications
Organize Purchasing, Traffic, and Receiving
Develop Despatching and Scheduling
Establish Order Progress Records
Determine Departmental Loads
Determine Limits and Tolerances
Analyze Rejections and Spoilage
Establish Quality Control
Salvage Surplus Materials
Utilize By-Products
Reduce Material Inventories

Take Physical Material Inventories
Improve Material Handling
Determine Causes of Cost Variations
Budget Plant Expenditures
Determine Payroll Distribution
Establish Standing Expense Orders
Determine Burden Rates
Set Machine Rates
Equalize and Budget Burden Expenses
Prepare Economic Reports
Prepare Market Analyses
Institute Research.

Labor

Organize Employment
Establish Training Methods
Prepare Job Specifications for Hiring
Prescribe Trade and Vocational Tests
Determine Basic Wage Rates
Install Wage-Payment Methods
Establish Personnel-Efficiency Records
Establish Man-Production Records
Establish Timekeeping Methods
Take Time Studies
Make Motion Studies
Analyze Operations and Jobs
Determine Operations and Sequence
Establish Standard Manufacturing Times.

Equipment

Survey Plant Location, Design and Construction
Develop Equipment Inventories
Analyze and Arrange Equipment Layout
Standardize Equipment
Improve Use of Equipment
Establish Machine Production Records
Standardize Cutting Tools
Develop Small Tools
Provide Tool Storage and Service
Improve Transmission and Application of Power
Improve Sanitary and Hygienic Equipment
Improve Means of Lighting.

APPENDIX NO. 2

QUESTIONS CONCERNING MANAGEMENT MECHANISMS

48 The questions in the Waste in Industry questionnaire, whose answers supplied information to prepare Tables 1, 3, 4, and 5, dealing with the use of the mechanisms of management, are as follows:

K7 Are your workers selected carefully on basis of interviews, trade and other tests?

K13 How is wage remuneration determined, union scale, competitive market, etc.; day work, piece work, or other form of incentive wage?

U4 Is your planning of work centralized or distributed among a number of shop executives and workers?

U9 Do you compare production performance with production standards, with reference to worker, job, department and equipment?

U5 Does your shop administrative mechanism enable you to anticipate idleness of machines and workers and provide against it?

U7 Have you a good current cost system tied in with the financial books?

U8 Do you compile a record of idle workers' time by amount, cost and causes?

U3 Do you compile a record of idle machine time by amount, cost and causes?

U11 To what extent is purchasing done to standard specifications, and does it clear through a purchasing agent?

U12 What control exists over receipt, issuance, and return of materials?

49 The questions in the condensed questionnaire whose answers supplied information to prepare Table 2, dealing with the use of the mechanisms of management, are as follows:

4 Storeskeeping: (*a*) General methods? (*b*) How are stock records maintained up to date? (*c*) Where are storesrooms located with reference to producing departments? (*d*) How is material transported?

10 Employment Methods: (*a*) What examination or tests are used? (*b*) How are discharges or quits handled?

13 Wage Scale: (*a*) By classes of labor? Ratio of day workers to piece workers? Is any form of incentive system used?

18 Production and Cost Control: (*a*) Is there a budget by departments? (*b*) Are periodic comparisons made between output by departments or trades and labor costs? Is overhead standardized periodically, or is *all* overhead distributed monthly?

19 Planning Functions: (*a*) Is there a central planning department, or does each department do its own planning and follow-up of

schedules? (*b*) How are schedules and standards set? (*c*) Are work tickets given out ahead of workers' needs?

22 Idleness: (*a*) Are records kept of idle machines and causes? Of idle men?

APPENDIX NO. 3

THE TAYLOR SOCIETY

50 This brief statement of the founding and development of The Taylor Society (originally the Society to Promote the Science of Management) was prepared by Dr. H. S. Person, the Managing Director of the Society.

51 *The Background.* In 1886 Henry R. Towne presented the paper The Engineer as Economist before the A.S.M.E. Thereupon followed in that society a series of papers relating to management, chiefly on wage systems, ending with Taylor's A Piece Rate System in 1895. Then followed seven years of no papers on management before the A.S.M.E. In 1903 and 1904 several papers were presented, among them Taylor's Shop Management. In 1906 Taylor's On the Art of Cutting Metals was presented before the A.S.M.E., followed by one paper on a phase of management in each of the years 1907, 1908, 1909, and 1910. In 1911 the A.S.M.E. had no paper on a management subject.

52 The group of young engineers associated with Mr. Taylor felt during this period that it required a struggle to get a paper on management before the A.S.M.E.; that each time a paper of significance was presented (Towne's in 1886, Taylor's in 1895 and 1903) consideration of the subject was stimulated, but the interest soon waned; that the dominant group in the Society did not believe management subjects should engage its attention and put obstacles in the way of such attention; that there was in the Society possibly hostility to the Taylor theories and methods and that they could not receive adequate consideration; and that in general the A.S.M.E. was not giving adequate attention to the subject of management as compared with the growing public interest.

53 On the following page is a tabulation of the number of items relating to management by years, and whether presented before the A.S.M.E. It has been prepared from *Scientific Management;* a list of references in the New York Public Library, compiled by Walter D. Brown, Technology Division, N. Y. Public Library, 1917.

54 This group therefore decided that to secure a discussion of management problems to an extent warranted by the importance of the subject and public interest in it, a forum other than the A.S.M.E. would have to be found.

55 *The Founding of the Society to Promote the Science of Management.* On November 11, 1910, a meeting was held at the Athletic Club, New York, to consider the matter. There were present Morris L. Cooke, Frank B. Gilbreth, Robert Kent, Conrad Lauer (representing Charles Day), and Wilfred Lewis. Mr. Gilbreth was the host.

56 It was decided to organize a society for the discussion and promotion of scientific management. A formal organization was not effected, but

from then on James M. Dodge presided at meetings and Robert Kent acted as secretary-treasurer. For two years, with such informal organization, meetings were held approximately once each month, usually at Keene's Chop House, where management subjects of live interest were discussed. The membership during this period increased to some twenty-five or thirty.

57 As a result of the increasing public interest resulting from the Eastern Rate Case Hearings (winter of 1911–1912), it was decided to make the organization more formal and to make more of the society. Accordingly a meeting was held at the Hotel Astor, November 7, 1912, and a formal organization effected. The Society was named The Society to Promote the Science of Management. James M. Dodge was elected president and Robert Kent, secretary. Meetings were thereafter held less frequently (three times a year) but were more carefully planned. The place of meeting was usually

	Year	Presented before A. S. M. E.	Not presented before A. S. M. E. Published in periodicals such as *Engineering Magazine* and others
Towne's The Engineer as Economist	1886	2	0
	1889	1	0
	1891	1	0
	1893	1	0
Taylor's A Piece-Rate System	1895	1	1
	1896	0	1
	1897	0	4
	1898	0	2
	1899	0	10
	1900	0	12
	1901	0	24
	1902	0	17
Taylor's Shop Management	1903	5	20
Taylor's On the Art of Cutting Metals	1904	1	22
	1905	0	17
	1906	3	17
	1907	1	22
	1908	1	46
	1909	1	30
	1910	1	57
	1911	0	219
Committee Report, The Present State of the Art of Industrial Management	1912	1	165

New York, Philadelphia, or Boston. In 1913 H. S. Person was elected president, and succeeded himself annually until 1919. In December, 1914, was begun the publication of a small journal for members called Bulletin of the Society to Promote the Science of Management. By 1917, when the United States had entered the war, the membership had increased to about 110.

58 *During the War.* During the war the activities of the Society were

in abeyance, the officers and over fifty per cent of the members of the society having been absorbed into the war organization of the United States.

59 *Reorganization and Change of Name to Taylor Society.* Immediately after the armistice in 1918, members of the society in Washington and the vicinity held a meeting to consider the resumption of activities of the society. It was felt that the society should undertake more serious work, in view of the probable larger public service possible during reconstruction, and it was decided to establish a central office with a salaried executive. The name of the society had been changed in 1916 to Taylor Society, in honor of Frederick W. Taylor, who had died in 1915; an office was established April 1, 1919, in the Engineering Societies Building, New York; H. S. Person was chosen the managing director, and John Otterson, Winchester Repeating Arms Co., was elected president. Mr. Otterson was succeeded by Henry S. Dennison, and Mr. Dennison by Richard A. Feiss, who is president at the time of this writing, August, 1922.

60 The objects of the Socitey as stated in the Constitution are, through research, discussion, publication and other appropriate means:

1. To secure an understanding and intelligent direction of the principles governing organized effort, for the accomplishment of industrial and other social purposes for the mutual benefit of
   - *a* The Community
   - *b* Labor
   - *c* The Manager
   - *d* The Employer
2. To secure the gradual elimination of unnecessary effort and of unduly burdensome toil in the accomplishment of the work of the world.
3. To promote the scientific study and teaching of the principles governing organized effort, and of the mechanisms of their adaption and application under varying and changing conditions.
4. To promote general recognition of the fact that the evaluation and application of these principles and mechanisms are the mutual concern of the community, labor, the manager and the employer.
5. To inspire in labor, manager, and employer a constant adherence to the highest ethical conception of their individual and collective social responsibility.

APPENDIX NO. 4

THE NATIONAL ASSOCIATION OF CORPORATION TRAINING

61 The National Association of Corporation Schools was organized on January 24, 1913, to formulate a definite and constructive educational program for firms engaged in industry and commerce. Its officers and directors were drawn from representative firms including: The New York Edison Company; Burroughs Adding Machine Company; General Electric Company; The Pennsylvania Railroad Company; The Curtis Publishing Company; Yale and Towne Mfg. Company; Consolidated Gas Company of

N. Y.; Dodge Manufacturing Company; National Cash Register Company; and the Westinghouse Electric and Manufacturing Company.

62 At the first meeting of the Executive Committee, an Educational Committee was appointed "to devise courses and recommend best how to teach: salesmanship, advertising, manufacturing, transportation, accounting, financing, purchasing, general office work, stenography, clerical work, filing, correspondence, physical efficiency, hygiene, sanitation, recreation, exercise and the elements of psychology. The general purpose was to assist firms having established educational work, or about to start such work. Through reports, members were informed as to what others were doing and what they might and should be doing.

63 For the first three years the emphasis of the work of the Association was directed toward preparing the new employee for his first working duties. In 1916 the scope was enlarged to develop employees old in point of service. By 1917 the work had broadened to include "all of the activities classified as human relations." This meant that the Association was definitely in the personnel field. In 1919 it was declared that the Association "has become a great clearing house for all authentic information on the subject of Employee Relations in Industry."

64 In 1920 it was believed that the Association faced a crisis brought on by the increased service being rendered and rising costs. A plan of reorganization was therefore developed providing for incorporation. The name of the organization was changed to The National Association of Corporation Training. The object was stated to be "the founding of an organization that shall contribute in every way possible to the mutual benefits of all concerned in industry and commercial enterprises; to develop the efficiency of the individual employee and to coördinate his best interests with those of employers; to develop the highest standards of efficiency in industrial operations; to have the courses in established educational institutions expanded to meet more fully the needs of industry and commerce; and to encourage all branches of literature, science and art, or any of them that pertain to industry and commerce." Three classes of members were provided for:

A — Commercial, industrial, transportational, financial, or governmental organizations

B — Any employee of a Class-A member

C — Individuals not eligible as a representative of a Class-A member, or as a Class-B member.

65 However, this reorganization did not modify appreciably the work of the Association, which continued until May 20, 1922, when the plan of merger with the Industrial Relations Association of America into the National Personnel Association was approved.

APPENDIX NO. 5

THE SOCIETY OF INDUSTRIAL ENGINEERS

66 In May, 1917, the Western Efficiency Society held a national convention in Chicago on The Importance of the Human Factor in Industry. On the day following the close of this convention, a group of engineers and executives met to discuss the human factor in preparedness and to consider the part which the expert could play in winning the war. At this meeting the Society of Industrial Engineers was organized. Before this meeting, the Council of National Defense had strongly urged the organization of such a national society and had indicated how such a body could assist the Government in the emergency of war. In June, 1917, the chairman of the Aircraft Board invited the directors of the society to a conference in Washington. As a result the society was called upon to gather through its members information on the personnel, financial and industrial resources of firms capable of building aircraft parts. Later work was done for the Ordnance Bureau of the Army. Until the armistice, the activities of the society were devoted to furthering the carrying on of the war. Two-thirds of the first board of directors filled positions in Government service; a majority of the members were engaged in organization and production work incident to the war.

67 During this period the activities of the society were devoted to the first object for which it was founded: "To furnish a vehicle for bringing together in closer relationship persons who are actively engaged in promoting efficiency in business and for making the training and ability of such persons available in the emergency arising out of the present war."

68 At the close of the war the second object became the guide for society work:

"To furnish a medium for bringing out original contributions to the science of management.

"To provide an organization through which persons who are applying scientific methods to the solution of the problems of production and distribution may exchange views and coördinate their efforts.

"To coöperate with other societies.

"To codify and standardize professional principles and practice.

"To develop the professional standards of the industrial engineer.

"To promote efficient, energy-conserving management.

"To enhance the efficiency and prosperity of American industry."

69 In July, 1910, the board of directors put into effect a plan of functional organization which has continued in force. The membership (September, 1922) is 1032, divided into six classes: patron, professional industrial engineers; professional technical engineers and accountants; managing executives of commercial and industrial activities; educators in engineering economics, psychology and other lines associated with management; juniors and students.

APPENDIX NO. 6

THE INDUSTRIAL RELATIONS ASSOCIATION OF AMERICA

70 This brief statement of the founding and career of the Industrial Relations Association of America (originally the National Association of Employment Managers) was prepared by Mark M. Jones, the first secretary and a director of the organization.

71 Sporadic attempts to organize the relationship between manager and men, rendered impersonal by the division of labor and the introduction of automatic machinery, can be traced as far back as the 80's. At first the development of this management function was slow. Somewhat later the number of specialists in the personnel field increased. It was not made up entirely of persons specializing on employment management. The growing importance of the whole problem of industrial relations had much to do with the formation of this association and while the majority of the members were specialists in the personnel field, a large number were either in the general management field, or outside of industry but directly interested in industrial relations.

72 Between 1910 and 1917 other local organizations for the discussion of employment problems developed in New York, Philadelphia, Chicago, Pittsburgh, Cleveland, Newark, and Rochester.

73 The inter-city aspect and the beginning of the national movement dates back to 1914, when men responsible for hiring held a meeting in Minneapolis for the purpose of exchanging ideas. The success of this small meeting was such that it resulted in a second meeting in the same city in May, 1915, which was attended by persons from a much wider area.

74 In May, 1916, a general meeting was held in Boston, and in the same month of 1917 a conference at Philadelphia was attended by several hundred.

75 At this Philadelphia meeting it became apparent that some medium for coöperation, a clearing house for the experience of local groups, was needed, and a committee of ten was appointed to consider the advisability of forming a national association. The committee reported that it believed the time inopportune for such action but recommended that a national committee be created for the purpose of arranging an annual convention, as well as continuing the study of the desirability of a permanent national organization. Adoption of this recommendation resulted in the creation of the National Committee of Employment Managers' Associations.

76 A meeting was immediately planned for Cleveland, Ohio, to be held during the following year. The war situation, however, caused a change in plans. The National Committee arranged to hold the convention in Rochester, in May, 1918, in recognition of the pioneering work of Rochester University in graduating the first class of employment managers trained under a special course for the United States Government.

77 An attendance of over 800 men and women at the Rochester con-

vention was evidence of the need for a more formal national agency, and after considerable discussion it was voted to organize a national association. An organizing committee was elected for the purpose of so doing.

78 When the National Association got under way during the closing months of 1918 it was mainly an organization of organizations. Its control was in the hands of representatives of local associations, for while there were three other classes of members, the group members representing local employment managers' associations held the balance of power and determined the policies of the National Association of Employment Managers.

79 The task of the National Association when first organized was stated as follows:

1 To arrange and manage an annual convention
2 Issue a bulletin
3 Promote the organization of local employment managers' associations
4 Assist local employment managers' associations then in existence
5 Operate a free employment service which would assist specialists in the personnel field to secure positions
6 Establish and maintain a central clearing house for employment information
7 Conduct such research work and surveys within the employment field as might be approved by the Board.

80 The first convention after the formation of the National Association of Employment Managers was held at Cleveland, Ohio, in May, 1919. An attendance of 2000 was an indication of the stimulus to the movement provided by the war.

81 As the result of a widespread demand arising out of the widening scope of the employment manager, with the result that his activities comprehended many more functions than could be described through use of the word "employment," the name of the National Association was changed on March 1, 1920, to The Industrial Relations Association of America. It was under this name that the first peak in the movement was reached. The annual meeting in Chicago, in May, 1920, attended by 5000, was one of the very great national conventions of that period.

82 Soon after the 1920 meeting the change in the business situation was reflected in the personnel field and a decline in interest and support set in. As personnel work had in many cases been the last addition to specialized management functions, there was a widespread belief that it would be the first to be discontinued. Experience has since indicated that such apprehension was not well founded. There were but few cases where that policy was applied in retrenching or where intemperate action destroyed a sound piece of work. Personnel work was decreased greatly, but not often to the same extent as other management functions. Wherever a severe reduction took place it came more as a result of individual work than an inherent weakness in the idea or plan.

83 In the first discussions of a national organization there was present a small group which was anxious to decentralize the unit of association

membership to the utmost by placing membership on an individual basis. The majority did not look with favor upon this departure, and the concentration of power and control in a group membership was the result. The experience of the organization ultimately established the fact that the group basis could not be entirely satisfactory. It did not provide the foundation for coöperation which was necessary if the Association were to make a real contribution to the progress of its members. It required many months, in fact, several years, for the advocates of individual membership to prove their case. However, at the 1921 convention of the Industrial Relations Association of America, held in New York City in November, a committee, widely representative of the whole country, presented a report which strongly advocated reorganization on an individual basis. The problems of the individual members of the National Board of Directors were so numerous, however, that they had little opportunity to apply themselves to the problems of the association.

84 Finally, the Board of Directors recognized the need for a considerable alteration in the structure of the Association and decided that the way might be cleared for the most expeditious action if the Association were dissolved and an organization established along the lines suggested to the convention by the Reorganization Committee. The necessary steps to that end were taken and the Industrial Relations Association of America ceased functioning on December 31, 1921.

85 An organizing committee for the new association was then at work, and out of the whole situation a merger with the National Association of Corporation Training was arranged. This crystallized in the formation of the National Personnel Association in April, 1922.

APPENDIX NO. 7

THE MANAGEMENT DIVISION OF THE A.S.M.E.

86 The Management Division of The American Society of Mechanical Engineers was organized in July, 1920. In the first annual report, the following definition of management was given:

"Management is the art and science of preparing, organizing and directing human effort applied to control the forces and to utilize the materials of nature for the benefit of man."

87 Interpreting the thought of that definition into a program, the same report stated the purpose of the Division to be:

"Inasmuch as the problems of management are of the utmost complexity and difficulty, the Management Division of The American Society of Mechanical Engineers in seeking to render disinterested service therefore declares its purpose to be the formulation and declaration of the fundamentals of management, both regulative principles and accepted practice, and the dissemination of Management knowledge.

"In working toward this object, the Management Division can thus

not only be of service to the other Professional Divisions of the Society, to the individual members of the Society and to other societies of like aim, but also to all who are in responsible charge of human effort, and therefore, through them can benefit society at large.

"In carrying out such a broad purpose the activities of the Management Division will vary with changing need, thus no comprehensive listing can be made to cover the present or the future. It is only possible to suggest a few already in project, namely:

"The standardization of management terminology, units of measurement, the improvement and development of management education; the elimination of management wastes in industry; the elimination of unnecessary fatigue in industry and engineering; and lastly, management research."

88 Almost from the start, the membership of the Management Division has exceeded that of any other Professional Division. It now (September, 1922) numbers 1740.

APPENDIX NO. 8

THE NATIONAL PERSONNEL ASSOCIATION

89 The National Personnel Association was formed to take over the activities of the National Association of Corporation Training and the Industrial Relations Association of America. The possibility of such a union was discussed informally at a joint meeting of the Executives Club of New York and the New York Chapter of the National Association of Corporation Training, held on February 17, 1922. On the following day a letter of invitation was drafted and signed by 20 men who supported the suggestion of union, requesting attendance at a meeting to be held on March 9. On that day 34 persons were in attendance out of about 100 invited.

90 It was unanimously decided to form a national organization devoted to employment or personnel activities, provided the two existing organizations could be combined. A committee was appointed to consult with the officers of these two organizations and report a plan.

91 This committee reported on April 7, submitting a plan for the new association and providing for an organizing committee to put it into effect. This committee was appointed and met on the same day. Anticipating this action, the officers of the I.R.R.A. had secured authorization to enter the merger. An expression of opinion from Class-A members of the N.A.C.T. was overwhelmingly in favor of the union.

92 On April 21 the articles of incorporation were completed and signed, putting the union into effect and bringing the National Personnel Association into existence. Its purpose is:

"To advance the understanding of the principles, policies and methods of creating and maintaining satisfactory human relations within commerce and industry.

"1 By assisting administrative executives, those engaged in personnel work and others who are interested in problems of personnel administration

through providing opportunities for conferences, coöperative research, and exchange information among members.

"2 By studying the problems of personnel administration, including employment, training, development, health, employee service and coöperation.

"3 By assisting established educational and other institutions to interpret the personnel needs of commerce and industry by maintaining reciprocal relations with them."

93 Two kinds of members are provided for: individual and company. The membership is (September, 1922) 500 individuals and 120 companies.

DISCUSSION

Fred J. Miller. Mr. Alford's admirable paper is, I think, very comprehensive, especially as a review of what may be called the tangible or measurable advances that have been made in the past ten years within the field to which the paper relates. I think, however, that the most important progress that has been made, mostly within the ten-year period, cannot be measured nor weighed, for it is manifested mainly in a changed attitude of mind.

At a time, about twelve years ago, when I, as factory manager, came into an intimate and responsible relation with the leading exponent of the art of Scientific Management then in active practice, and was in daily contact with his work, the dominant idea was that a new system of management was to be handed to the body of factory executives; they were expected to learn by rote to do each little part in working the system, and to ask few if any questions as to the whys or wherefores. This did not go very well, especially with our large group of department foremen who had been accustomed to be consulted, to have their opinions more or less deferred to, and to be left pretty much with a free hand in the management of their departments. Some of them resisted passively; others of them somewhat actively; scarcely any of them really liked it, especially at the beginning. Questions of discipline arose and a considerable proportion of previously satisfactory minor factory executives would have been dismissed if the advice of the industrial engineer had been followed. Eventually this led to an agreement that all matters of discipline and of bringing factory men into line were to be left to the regular factory executives. This worked much better, and the installation was completed with entirely satisfactory results. But it was not until

we had entered the World War and the necessity arose for devising a method that would immediately, almost from the first day, without change of personnel, bring about an improvement in efficiency and promote smooth working of an organization already in being, that the difficulty was entirely overcome.

Just here, stimulated by patriotic motives, Mr. Gantt, assisted by his staff, did, in my opinion, his most important piece of work. He had the breadth of mind needed to perceive that he was confronted by a new set of conditions that, so to speak, compelled the enlistment and active participation of the foremen and others already familiar with the technique of the work they were engaged in. Many of the things formerly done by clerks were then done by the foremen or men working under their direction, — and much better done; but, better still, the foremen no longer felt that they were being shelved or sidetracked. On the contrary they realized that their status was improved in many ways. Thus the war led to Mr. Gantt's conversion to our view, that foremen and others in the factory ought to be more actively engaged and their interest enlisted in the new methods of administration.

Mr. Gantt rose to the occasion magnificently. He entirely abandoned his earlier method of approach in installing his work, and found, what most intelligent shop men already knew, that factory men respond quite as readily and heartily to considerate and fair treatment as do other people.

I look upon the development of a method which is compatible with the facts cited above and in harmony with them, and which promotes coöperation instead of arousing opposition and antagonism, as being the most important advance of the past ten years and as being at least one superlatively important result of our participation in the war. This is so for the reason that aside from the fact that it is the decent, human thing to do, a method by which we attract and secure the hearty coöperation of those whose coöperation is vital to complete success is, obviously, scarcely less important than improved management methods themselves. In the earlier days especially, there was far too little of real management in the installation of management methods; and their adoption was, — and for that matter, still is, — much retarded by that fact.

H. A. Person. I think Mr. Alford showed a great deal of commendable bravery in tackling this particular subject. Ten or

twelve years ago when the first report was made regarding progress in management, the problem was relatively simple, for the reason that there had been, during the preceding twenty years, marked advance in the development of management, and particularly marked advance in the statement of principles of management. Now, when one tackles the problem in 1922 of measuring the advance during ten years, over the achievements of 1912, one has a very much more difficult problem — in my judgment, almost an unsolvable problem; and, therefore, while I admire Mr. Alford's bravery in attaching this particular subject, I am not of the opinion that the achievement, in the very nature of the case, is comparable with that of ten years ago.

As I recall the printed paper, the first pages are devoted to proof on the basis of opinions. Now, I do not think much of proof on the basis of opinions. It is true that the report of ten years ago had a certain amount of proof on the basis of opinions, but it was a very much smaller proportion of the whole, so for a moment, I want to eliminate that.

Then, in the appendix and the latter part of this report is a very considerable amount of what I should call not exactly irrelevant matter, but matter which is over-stressed and does not seem particularly impressive as proof, in proportion to the amount of space devoted to it.

That leaves us what I may call the central part of the paper — the more substantial proof based upon certain data secured by the Committee on Elimination of Waste in Industry.

Now, with respect to that I have two particular observations to make: First, those data had been used once before by the Committee on Elimination of Waste in Industry to prove that industry is not so very efficient, from the point of view of management, and, of course, one has to be cautious in using it over again as a sort of proof that there has been very considerable advance in management. Nevertheless, in the second place, it does indicate an advance in management during the past ten years, but not in a positive, a precise and what I would call a measurable way.

We have, as evidence of advance in management, the existence of these various mechanisms which are tabulated in that part of Mr. Alford's report, but the mere existence of these mechanisms is not sufficient proof to me of a great increase in the art of management. It is simply proof that there has been some increase; for the excellence of management, in my judgment, is a result of these

elements in combination, and when one counts them and tabulates them as isolated phenomena, one does not get that measure of proof which one would get by counting them and tabulating them in combination, because one plus another means a very much greater advance in management than two times either one alone.

In my judgment, a real measure would be an analysis of the existence of these mechanisms in combination, and that I conceive to be an almost impossible task at present.

I think, as Mr. Miller has said, that our greatest measure of advance in management is that more or less intangible, but very real, thing which we all feel in the industrial atmosphere — a new point of view, a new approach — and, in my judgment, that is the big thing; so that while I cannot say that this paper compares in significance to that of ten years ago, I think Mr. Alford is to be commended and congratulated for attacking the problem.

J. P. Jordan. The question has been asked, "Why does the industrial or management engineer fail to have the standing of a mechanical, civil, electrical, mining, chemical or any other engineer of like class?" Is not this question pregnant with wide opportunity for much discussion? And can we not find in this paper by analysis a possible answer?

Par. 10*d* states "Acceptance of the principles of management has broadened among engineers, executives in industry and educators." May I ask what are these principles? Has any individual or any group of individuals fixed any such principles which have been generally accepted? and if so, who? Frankly I personally do not know of any such principles which can be accepted as such at one hundred per cent value, with the exception of two or three of very general import.

In Par. 15 we find in the arrangement of the eight "mechanisms" in the order of the number of plants where the installations were good, that cost control is seventh or next to the last. We then find in Par. 16 that cost control is again in the rear rank. This conclusion seems to me to be a most serious and damaging flaw in management progress, if it should be really true that such a condition exists generally in industry. Do we class general or financial accounting as a mechanism of management? Certainly not. Why, then, should we class cost control as a debatable mechanism rather than an absolutely indispensable organ of enlightened management?

Par. 19 states that an attitude seeking to make foremen and even workers conscious parties to the development of certain mechanisms has been noted, the effect of which is the installation of methods and mechanisms from the *bottom upward.* This is invariably so in plants where recognized and properly executed cost control is in effect; and it is not only a perfectly natural result, but probably it will be found that the upward demand for various so-called mechanisms occurs *only* in plants where cost control has clearly indicated the need for such mechanisms.

Under the general heading of "The Human Factor in Industry," Pars. 26 to 30 inclusive, I feel that Mr. Alford has restricted his consideration to but one phase of industry. The confining of thoughts respecting the human element to actual workers, to the exclusion of superintendents and foremen, seems to me to miss the biggest point of all. And this again leads us to cost control.

The first and greatest step is to humanize the foremen by giving them full and complete cost statements, — something which exists in comparatively few companies today. And a continual neglect of this feature will rob industry of a most powerful and necessary means of progress. The lack of such backing for foremen has probably been a leading reason for the handling of workers on a bluffing basis, with no definite facts to work on or show. Without such facts efficient work was subjected to the same pressure as the inefficient work which was actually causing high costs. Therefore, those workers who were unfairly borne down upon, resented it; and we deliberately created the conditions of labor calling for the works councils and other movements mentioned under "The Human Factor in Industry."

It is to be regretted that "management" has not awakened to the realization of the necessity to "Know Thyself"; that the advisers of management have failed absolutely as a class to recognize cost control as *the control;* that all such things as time study, incentive wage plans, planning, idle time analysis and such are but means to the end of reducing costs; and that management is so blind to the creed that facts and facts only should guide their every action. Then — when facts are known, and known month in and month out, put into operation such necessary adjuncts of management as may be clearly indicated as necessary.

I want to express my regrets to the Management Division of The American Society of Mechanical Engineers, the Society of In-

dustrial Engineers, the Taylor Society, and the National Personnel Association, that Mr. Alford did not include with you our little organization of about 3,000 members, called the National Association of Cost Accountants, of which I have the honor to be President, which is dealing with the problems of management along the lines as brought out in this discussion.

Perhaps we are wrong in our premise, but we believe in taking the position that the managements of industrial institutions are fully capable of fairly proper action, if they know, through accurate and usable costs where action is most needed. If figures show enormous costs in burdens for handling materials, management surely will cure it; if inventories are mounting on account of congestion in shop, management will surely take steps to use the standardized records and put in planning methods to reduce the congestion; and so on with all other mechanisms which may be indicated as necessary.

Management can progress only as it sees the light; and notwithstanding the apparent apathy of management, and perhaps also its professional advisers in the past, I feel that the future progress of management will come only through the guidance of cost control, accurately determined, properly presented, adequately interpreted and effectively used.

Dexter S. Kimball. My own view of this matter is somewhat different from those presented by the preceding speakers, perhaps for the reason that I was long in close touch with industry and later have had a chance to stand at one side and see it pass by.

The great idea that sprung from the mind of Mr. Taylor has often reminded me a great deal of religious growths. Some great prophet gets an inspiration and teaches, through parables, what he essays to get before men's minds. Mr. Taylor's great paper, which is the fountain-head and will probably remain so, of this movement described a specific form or piece of mechanism of management. Many interpretations, however, have been put upon this paper, just as many and diverse interpretations have invariably been made of the work of all religious prophets. One man finds in one particular item the thing that he considers the vital truth, while another pins his faith for salvation on an entirely different passage of Scripture.

The question then as to whether we have or have not made

progress cannot therefore be measured by the progress made in any one portion of the field, and I agree with Mr. Miller that it is very difficult to find criteria to tell us whether we have made any progress. This much is certain, however: the human race has never had before it for any length of time any great economic principle, and seriously considered and pondered over it, but that it has emerged with the truth. And out of the movement Mr. Taylor started there will surely emerge certain basic principles. Of course, some of the practices that Mr. Taylor advocated have already been discarded by forward-looking managers, but nevertheless as time goes on it becomes increasingly clear that this classic document contains certain important truths, basic and fundamental, that are destined to influence industrial management for many years to come. Mr. Alford's paper, if I interpret it correctly, is an effort to bring out more clearly to our view the most important results that are emerging from this movement.

It is not difficult, I believe, to recognize the most important of these basic fundamentals as such but it is not so easy always to see just how and where they should be applied. What is applicable to one shop is not applicable to another. The plan that may operate well with one set of men may not operate at all with a group having different personality. This is essentially true of cost keeping which is nothing more or less than one form of recorded experience so strongly urged by Mr. Taylor as a basic requirement for predicting results. Mr. Taylor did not invent cost-finding, however, though his work and writings no doubt gave an impetus to its development. As a matter of fact, cost-finding has been a matter of steady growth for many years. Not so long ago this science, like many others, was almost purely empirical in its character, and I have seen bids on great battleships worth at that time four million dollars prepared by a single individual with only such data as he himself had gathered during a busy life and contained mostly in personal note books and similar records. Today cost-finding forms a part of the education of most students of engineering.

This in general is true, also, of other economic industrial principles. Mr. Taylor himself disclaimed the invention of any new thing in management. In effect he said: Here are some things that are true: here is a combination of principles that will produce results, though no doubt there are other variations and combinations of these basic principles that will produce equally

good results. I consider Mr. Alford's paper, therefore, a very important one since it presents a clear résumé of the progress made in interpreting and applying Mr. Taylor's gospel of management.

I agree fully with Mr. Person that it is difficult to mark progress. As time goes on the curve flattens out and it becomes increasingly difficult to see what principle should be applied next to keep the movement progressing. But I do believe that as time goes on the fundamentals that underlie successful management will be made clearer and clearer and that these laws will be as well known as those that underlie engineering or any other well founded science. I have every confidence that the industrial engineer and the cost accountant will eventually come into his own. Not the least important index that this is true is found in the fact that every progressive educator believes that no man should leave college without some training in economics and in the fundamentals of industrial engineering and all good technical schools are now teaching these fundamentals as they conceive them to be at present.

R. L. Sackett. Dean Kimball has commented on the educational feature and I want to point out two evidences of progress in management, both of which are educational:

Nothing has been more striking than the sudden increase in foreman's training, which ten years ago did not exist as an organized educational movement. Those who have been in touch with industry know that no educational movement was ever introduced into industry until it had gained the sanction of administrative officials. Foremen do not take up educational work; no man in the ranks does until it has had something more than the tacit support or encouragement of those who are responsible for management.

The other evidence is that there has been a reflex in engineering education, and there will be a more marked reflex. Those institutions which have installed courses in administrative engineering or industrial engineering have taken the fundamentals, as Dean Kimball has pointed out, and built upon those a course; and their minds have been directed more toward the personal relation than they had been directed in the old courses, which gave more attention to materials and their uses. My point is that the development of management has been accompanied by the development of those courses, and those courses are going to have, and are

having influence upon the courses in civil, mechanical, and electrical engineering in drawing the attention of students, during perhaps the senior portion of their career, to the humanistic and personal side of industrial engineering.

David B. Rushmore. In the number of points that are open for discussion on this paper I wish to add only one thought, and that is that in management there is need of a definition as to what management is. Broadly, I conceive it to be a function connected with industrial organization, but we need a definition as to what organization is. It is something made up, in part, of human beings, and we certainly need more or less, a definition in this work as to what a human being is, as to what he is composed of, as to the motives which actuate him, as the results of certain forces that play upon him.

If we do as Professor Kimball said, — enunciate the fundamental underlying principles, — we are doing the best we can in any line of work.

It has been my experience to see in a large number of organizations men attempting to function as executives without having devoted any thought of their own as to what an executive is. I am very sure that Mr. Alford and this committee will accomplish a great deal in bringing to the attention of the younger men who are going from the colleges into industry the suggestion that they acquire a clear appreciation of what an executive is before they undertake to function as executives.

Some of our young men, the moment they get a stenographer, become executives. We find a great many men in industry, large and small, who are not making the most of themselves, and many remain in the same positions much longer than they ought. In general we blame the man, but to some others it seems that where a man is not getting the best out of himself, the man to whom he reports — his executive — is, in part at least, responsible. One of the great needs of industry is a clear vision on the part of the younger men of the province of an executive, the functions of which are rapidly changing; as to what the possibilities are, what the responsibilities are, what the opportunities for accomplishment are, and what the line of future development should be, along the line of these fundamental principles for making the best executives. A large number of men who are trying to function in this way are, as I see it, many of them, doing it without any analysis, with-

out any thought of their own, and doing it in about as many different ways and acting on about as many different principles as there are individuals; and this committee will indeed do a great work if it can inject into this situation a sense of the desirability of the development of executives along fundamental lines, which will be a real constructive work for industry.

L. W. Wallace. I feel sincerely that we owe a debt of gratitude to Mr. Alford for having searched all sources of information for the purpose of presenting before us the significant developments in relation to management during the last ten years.

I know something of the task of trying to correlate information, because one of the shortcomings of management today is that it has not kept systematic and fundamental facts and information on record and made them available to a large number of our people. I think the important thing that we need to do, from a management standpoint, is to make available to a larger number of people, and especially to employees, the fundamental economic facts regarding American industry. We, as industrial leaders, if I may be so immodest as to use that expression, for the moment — have been too content, we have been too complacent, we have sat behind our desks and let our employees get their information regarding the economics of industry from the speaker on the soap box in a street corner; and it is time for us to begin to open our eyes, and, as Mr. Alford has said, to begin to train from the bottom up, instead of from the top down; so I think that, from the standpoint of our employees, we should begin to give them the fundamental information that they need in order to arrive at a fair judgment as to what industry means to them; in other words, and in brief, I am arguing for greater publicity, the opening up of our books and our cost records to our employees, so that they will realize that the cost of production involves something more than simply wages and material. If industry is to go forward without being hampered by federal and state legislation and by various commission forms of regulation, it is going to be necessary also for management, in the coming years, to give to the public the fundamental information that it should have and is entitled to, regarding industry.

So I hope that within the next ten years we will see a decided step in the direction of giving to employees and to the public fundamental information regarding industrial economics; and that pre-

supposes that the leaders of industry will have to collect and codify this information for themselves. In our investigation of Waste in Industry we found that in numbers of instances, if not in almost every case, there was a serious lack of fundamental data within the organization itself. I may say to you that I am rather inclined to believe that many managers are more ignorant regarding the fundamental economics of their business than many of their employees. Before management can intelligently give to labor and to the public the information that it deserves and should have, by right, management itself has got to become more informed as to the fundamental economics involved, and I hope that the next ten years will show a marked progress in that direction.

Frank B. Gilbreth and L. M. Gilbreth.[1] Mr. Alford's plan of basing the progress report on the report of the Committee of 1912, of which he was a valued member, has both advantages and disadvantages. It furnishes a definite method of attack. On the other hand it restricts observation. New developments may be omitted or, if observed and recorded, emphasis may be incorrectly distributed, thus both recording and weighing of data may be affected. It necessitates a thorough review of the 1912 Report, with its discussions and closure, before the new report can be adequately criticised.

The four points specially emphasized in the 1912 Report are:

1 The advance in unskilled work and the trades. (Par. 1) "The unquestionable proof of the advance that can be made in unskilled work . . . and in ancient trades. . . . These are the most striking phenomena of all."

2 The change in mental attitude toward the problems of production (Par. 22) which is called "the most important change and one that comprehends the others."

3 The transference of skill (Par. 41). "We have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry, is the transference of skill."

4 The making of utility the aim of laboratory methods (Par. 56). "The end and aim is utility."

This being the basis of the 1912 Report, — and a careful study will make this clear, we may expect to find these four points extensively discussed, — in fact made the basis of the 1922 Report.

What do we find when we turn to the 1922 Report? It starts

[1] Very much abridged.

with an account of the limitations of the report; states its methods; notes the loss of three great pioneers and outlines the method of acquiring data. It then states as the chief subject the "new element," "the mental attitude that consciously applies the transference of skill to all the activities of industry." Then follow quotations from answers to the questionary as to progress since 1912, deductions from these plus "certain well-recognized facts," a review of progress in use of management mechanisms, a summary of management education, a statement of the results which affect the human element, of the acceptance of the "service motive," and of results that affect production and ultimately industry, business, community, state and nation.

The only strong emphasis noted is on consideration of the human factor. This is one of three factors introduced as concerning "especially significant developments, which after being stated are subject to explanation and comment" — and it is stated later that "appreciation of the importance of the human factor in industry and attempts at its study from a fact basis have been the most striking management development."

The questionary method is always open to objections — as those who have used the questionary, or are acquainted with its use in the field of education well know. It is difficult to select those to whom to send the questions; only a small per cent respond; and only a small per cent of responses are of value.

In examining the four factors of application of the 1922 Report (Par. 11), we note that they have to do largely with the keeping of records, — cost accounting, specifications, graphics; also with the extent to which management has been introduced. We find no list of measurement units, measurement methods or measurement devices introduced or developed. Yet we find stressed "the demand for knowledge, facts, as a basis for judgment."

Among the significant factors (Par. 12) it is well stated that management has expanded beyond the mechanism of the Taylor System. It might be appropriate also to list those mechanisms of the Taylor System which, while obviously antique, obsolete and superseded, are still in use — telling where, by whom, and especially why.[1]

In the review of the college courses, the nature of the work as it concerns technic might well be mentioned.

The study of the human factor "from a fact basis" has gone

[1] June 1921 Taylor Bulletin.

far beyond "attempts" (Par. 12k). Detailed discussion should be invited on this point.

In the discussion of management mechanisms that follows (Par. 13) the consensus of opinion seems to be that these are generally accepted in principle, but not in practice. The final sentence of Par. 13 states admirably the reason why this is the case.

The tables discussed (Par. 14) are interesting, but the vital point is — *why* are the results as they appear to be? Why is Time Study, for example, sixth, seventh and eighth on the lists?

The 1912 Report stressed the human element. The 1922 Report states that the human factor is today the important element — yet the report places only five short paragraphs (Pars. 26–30) under this topic. This is not because Mr. Alford underrates the importance of the subject, but because he reflects the customary failure to realize that *training the worker is the real key to industrial relations.* The work of an employment department (Par. 27) and of "works council" (Par. 29) is not to be disparaged, but the *training of the worker,* instead of being dismissed in a few phrases in a paragraph or two, should cover that transference of skill which is stressed in the 1912 Report.

It is necessary to note most carefully the results of progress listed (Pars. 42-46). The only new result claimed since 1912 is the reduction of cost of the product to the consumer.

It is claimed that, through the work of "generations of effort" management now has "a great body of knowledge and practice," by which it directs both planning and performing.

But has skill been transferred during this period? If so, to what extent? If not transferred, has it been recorded? Are the data available to be formulated into transferable form for transference later If not, why not? The report fails to say.

Real progress has been made. Skill can now be, and is being recorded, and is being transferred. The demands of those who wrote and discussed the 1912 paper have been met to an extent sufficient to prove the wisdom of that report.

The appendices are valuable as bringing much material into brief reviewable form.

A most unfortunate and serious error occurs in Appendix No. 3,[1] where an account is given of the founding of the Taylor Society (Pars. 52–57). As the writers of this discussion thought of and planned this first Society of Management, selected and invited

[1] Contributed by Dr. H. S. Person.

those who were to attend a dinner which was the first meeting of any Management Society, and then and there disclosed and outlined the purpose of the gathering, we may therefore speak with authority.

We can state without question that the Taylor Society was not founded because of dissatisfaction with the attitude of The American Society of Mechanical Engineers toward management. The idea in mind in calling the group together to form the Society was to furnish an opportunity for men intensively interested in the principles and practice of Scientific Management and occupied in installing such management to discuss not only the theories and philosophy of management, but the technic. It was to furnish opportunities for the exchange of experience and for best utilizing such time as Mr. Taylor had to give those interested in the new type of management. There was no feeling of criticism toward The American Society of Mechanical Engineers; no desire to withdraw from the Society, in any way or in any sense, and, so far as we know, all members of the original group were, always have been and are at present enthusiastic and loyal members of The American Society of Mechanical Engineers.

We have noted that the keynote of the 1912 Report was the transference of skill. This idea was not new, as Mr. Alford himself acknowledges, but it was very well phrased, and its appreciation, and emphasis in the Report mark a milestone.

Too much emphasis can hardly be laid on the transference of skill, because the further developments during the last ten years have proved beyond doubt the importance of emphasizing everything that has to do with the most efficient transference of skill with the least loss in transmission. Naturally, to transfer skill it must first be recognized, recorded, and put in such form that other people than those who *have* the skill, or *notice* the skill, or have the ability, facilities and opportunity to record the skill can actually leisurely refer to, visualize, measure and evaluate the data that contain the skill.

This transference of skill does not mean the transference of skill of a laborer or mechanic only, — it also means the skill of the manager, the executive, the artist and everyone from the super-expert, to the novice of all classes. There is skill in performing manual operations, and there is skill in performing managerial functions and operations, the laws of acquiring skill and automaticity in manual laborer's work and that of the mental laborer are

identical, even though the results and their behavior are entirely different.

The transference of skill is a very great conception, as Mr. Alford realizes, and obviously, if we are to emphasize the transference of skill we must observe those who are the most skillful.

There must be great emphasis placed on the type of method which is used for recording, otherwise "transference of skill" becomes mere words, as it often has.

There will be no transference of skill until the method of work is recorded *in detail.* This is a fact not mentioned in the 1922 Report.

The recording of the method in measured errorless detail is the step precedent to any attempt at the transference of skill. Because of the lack of recording of the method in detail, Taylor was obliged to discard all of his time study data and the data of his associates when he started planning his book on time study which was finally written after his death by Dwight Merrick.

The state of the art in 1912 can be seen where it is advocated that the end and aim of management investigation be utility, and consideration of any other purpose than immediate utility is deprecated. There is no comment on this in the 1922 Report. The fear, in 1912, was that there would be too much time study and motion study. There was no recognition at the time that, inasmuch as all accurate motion study records not merely so-called elementary unit times, but actually the elements of the motions themselves, therefore all observations are absolutely transferable to other work immediately, besides being of the same value for all future work. This answers any possible criticism of the management investigator using his laboratory methods to discover facts apparently other than those for immediate use. All knowledge of any trades that has to do with elements of motions is *usable in all other trades,* for the reason that information regarding each of the sixteen subdivisions of a cycle of motions is usable in every kind of cycle of motions.

On every hand are found words emphasizing the training of the worker, but where and in how many instances are found those who train the workers, knowing in detail how the work should be done, and how they should be trained? There is but one thing that will hold the respect and the attention of the worker permanently, and that is training in the one best way to do work, based on measured facts together with original data that he can look over

just as an engineer can look over the derivation and source of a formula given to him.

The respect of the worker will never be obtained when data are accumulated only on the fastest or most strenuous or most continuous worker, instead of the worker with the best method. No one thing has done so much to kill progress in scientific management among the workers as the misplaced emphasis on the time it takes to do work, instead of the one best way to do work. Usually the one best way to do work will produce the quickest time, but this is not necessarily so.

We desire to emphasize the fact not as yet generally recognized, that any information regarding methods where the motions are recorded in their most elementary details is usable for synthesis in making the one best way in any other kind of work, and also permits the prophecy of times, even when automaticity in the new synthesized method has not yet been acquired and is not yet in existence, and therefore cannot be actually demonstrated in practice. This is of vital importance, for time study must include the prophecy of the time it will take to do work after the worker has become highly skilled, not merely the time the work did take during the timing period.

It must never be forgotten that the quest for the one best way and the search for records of skill worthy to be transferred have the same object in view: The former simply adds the thought that the skill shall embody the best elements of method to be found, thus offering a method to be transferred that may excel any skilled method in actual existence. The one best way is not only an adequate answer to the demand of the 1912 paper for records of skill and their transference, it gives more than was there demanded.

We maintain that stop watch time study, although much better than nothing, will not give information on details of method. These can be obtained only through details of motions, which must, therefore, be completely separated from time study both in description and definition.

The realization that method, consequently detailed motions, must be considered, studied and recorded, placed new emphasis on the importance of considering superstandardized equipment and tools. An important new condition of the new type of management is the emphasis on furnishing the best tools procurable to the worker, instead of having the worker use such tools as he may de-

sire, or he may provide for himself. This is important under all types of management, and is stressed under scientific management, but it is vital if the one best way is to be used and maintained, and the resulting skill to be utilized.

It may focalize attention upon distinct and definite advances since 1912 to list some which are the direct result of emphasis on the one best way to do work, as follows:

1 The recognition that management consists of definite fundamental units and that the greatest progress in management can come only by the selection of the right units to measure, the right methods for measuring these units, and devices that will make the cost of measuring the units accurate and cheap.

2 The recognition that the sixteen elements of a cycle of motions and not the motions are the true fundamental units for the one best way to do work.

3 Process charts and other devices that show the sequence and relation of operations of making, inspecting and moving, in a form that makes possible easy visualizing, as a whole and in detail, of the problems confronting the management in the plant, the office and the sales or any other department.

4 Instruction sheets for operations based on accurate knowledge of the one best way to do work.

5 Graphical control on the exception principle for executives, with provision for those executives to write and sign causes of deviation from class, task or program.

6 Standing orders for executives as well as others, with provision for notification, inspection and enforcement, together with the reason for the standing order, that all members of the organization may coöperate intelligently. This results in one best way maintenance.

7 Change orders with standard provisions for changes in the procedure and practice of management, together with the reasons for the change.

8 Suggestion systems and the proper accompanying mechanism for obtaining, absorbing and enforcing the benefits of the craft and managerial knowledge and skill of all of the individual members of the organization.

9 Superstandardization that will permit working in the one best way for the highest wages combined with the lowest costs.

10 Recognition that automaticity resulting from standardization is the greatest free asset of the worker.

11 Recognition of the mnemonic classification as a necessity for the filing of information regarding the one best way to do work.

12 Recognition that data on skill for current use only, is an unnecessary economic waste, and that, for maintenance, records must be in such form that, once made, they are usable forever.

13 Recognition that for recording skill, it is an economic waste to observe any other than the best men obtainable.

14 The application of and conscious and intentional use of psychology to solve the problems of management.

15 Devices for obtaining, synthesizing and transferring the precious skill of super-experts.

16 Devices for an efficient learning process for all, including those who have neither the opportunities, technical knowledge or apparatus to record or to derive the one best way to do work.

17 Provision for each member of the organization to share in the management according to his ability.

18 The recognition that the percentage of labor turnover is a measure of the efficiency of the management.

19 The recognition of the economic value of the placing of all workers, including the handicapped worker, in the highest positions that they can fill.

20 Standard opportunities for fitting for promotion that maintain stability and at the same time recognize individual aptitudes.

21 The recognition that the waste from unnecessary fatigue is as definite and as real as the waste of tangible material. The organization of the Fatigue Committee of the Society of Industrial Engineers to attack the problem of the elimination of all forms and instances of unnecessary fatigue as an economic and social duty as well as a philanthropic act, and to disseminate the information that any fatigue elimination means more output or more comfort, or both, and that fatigue study is a first step in motion study.

22 The recognition of accident prevention and the safety first movement as a managerial duty.

23 The recognition of the value of professional management engineers as executives.

24 The recognition that all blank forms that have not been observed in accordance with the laws of motion study are entirely obsolete.

The above twenty-four advances in management have been either entirely omitted or merely hinted at in Mr. Alford's report.

They are the memorable fundamentals on which management has progressed. They will be the fundamentals on which management will progress, because they are economically sound and conform to the laws of psychology, the greatest law of which is the Golden Rule.

Wilson P. Hunt. In discussing the question of management and its advance and improvement in the past ten years, I naturally lean towards the human problems as my training and experience have brought me in constant touch with this side of the question, having been brought up in the shop and still taking a special delight in keeping close to the shop side of things and the men in it. There is no doubt that tremendous advances have been made in the past ten years along the lines of efficient and scientific management. After Taylor and his associates showed what can be accomplished by time and motion study, many so-called efficiency experts broke loose and what they did in the name of efficiency and scientific management was a wonder.

It seems to me that one great weakness of most of the time study and efficiency plans has been the assumption that all of the brains were in the office, entirely ignoring the vast store of experience and knowledge among the men in the shop. A man who proceeds on this plan and forces his schemes and plans ready-made on the men in the shop without seeking their coöperation is missing a lot. Any plan that does not seek the coöperation of the workmen and get them to *want* the improvements and changes in place of forcing them may be made to work, but it will not accomplish one-half of what might be accomplished by proceeding along the other line. The handling of men, recognizing the human element in all dealings is the big problem of management, whether it be in the buying, the production or the sales end. The tendency is growing towards having the employees more directly interested either through some form of stock-holding or profit sharing, or through the various forms of work councils. I believe along these lines that the greatest advance is to eliminate the friction between employer and employee, as the farther we go along these lines, the more certainly is the employer an employee and the employee an employer, and he cannot very long strike against himself, nor can he help to tread himself under foot. I believe the next ten years will show wonderful progress along these lines.

H. A. Soverhill. My experience tells me that it is necessary to have sufficient management installed in a plant to give results.

It seems necessary to have a sales or shipping schedule, stores records, bills of material and cost system at least sufficient to give the desired results.

It is very evident that the same management mechanisms do not fit a jobbing shop and a strictly manufacturing plant. The underlying principles are of course very similar.

The name Scientific Management has left a long string of disappointments and I quite like Mr. Alford's reference to a term we can call Common Sense Management. This sort of management applied in the proper manner will surely produce a good working organization and results that will be satisfactory to the stockholders.

Max Sklovsky. It is well at the start to dispose of the skepticism that has existed on the subject of scientific management. As in the medical profession, there has developed what might be called quack engineering, and patent medicines have from time to time been prescribed by so-called engineers, one of them having as many as twelve patents which he has seen fit to classify under the dignified term of principles of efficiency. Mr. Alford very wisely ignores all of these and adheres strictly to the legitimate endeavor.

It is no reflection to recognize that many of the engineers that followed in the trail of such pioneers as Frederick Taylor and Gantt have failed in the fulfillment of their claims. Their chief weakness lay in:

a Too great a generalization
b A lack of appreciation of "inertia" in human enterprises
c A lack of appreciation in the human element
d Lack of understanding of executive viewpoint
e Lack of knowledge of selling methods.

The result of the work of these men has been that many industries have started on a series of experiments in methods and management, resulting in a series of disappointments in many instances with disastrous results. These disappointments were the result of assuming the presentation of the engineer as a practical plan, and the failure to materialize had, therefore, cast a reflection upon so-called scientific management. In the long run, however, it may be said that these experiments have been a fore-runner of real progress that has since been and is further to be made, so that industry as a whole, has gained as a result of the efforts of these

men notwithstanding the many blunders and disappointments resulting in many cases of a far-fetched and unwarranted claim. The engineering profession will, however, live down these mistakes and possibly some day hail as heroes and benefactors the men that we have in more recent years whole-heartedly discredited as false prophets. The chief error on the part of these engineers has been that they endeavored to reconstruct industry quickly by revolutionary methods in place of recognizing that the slower evolutionary method was the path by which the greatest progress would be made. This haste in the endeavor to establish new conditions has resulted in the use of undigested and unreliable data with the result in damage as followed. It is no surprise, therefore, to note in Mr. Alford's paper under Item 8 that opinions on progress in management vary widely and also that in cases reaction had set in. This, however, is a temporary attitude which will correct itself by time. There is a healthy index in the endeavor on the part of engineers through special study courses to attain a better understanding of business enterprise. The engineer has been usually weak in this direction. His business equipment had been very close to the vanishing point and as a consequence he has ignored the selling side, the financial side, and the human side of business enterprise. This attitude on the part of the engineer is shown to be itself weakest in dealing with intangibles, particularly with the human factor. It is no secret that on the whole, men without formal training have succeeded better in dealing with men than the engineer, but the engineer is waking up.

The data given by Mr. Alford in the tables indicate conclusively that *control* of industry has made substantial progress in dealing with the physical side of the business, but comparatively little progress has been made in dealing with the intangible or human side. Just what progress may be made in the future in that direction is difficult to predict. In fact it appears that it may be wisest for the engineer to leave that field entirely alone and apply himself to the development of the physical in industry, holding at all times in mind, however, that the human element is a factor that should always be taken into account and that to a large degree it is an uncontrollable item.

The Author. Ten years ago, when Mr. Dodge gave the closure for the Committee, which reported on the status of management, he said substantially this: It was unnecessary for the

Committee to make any closure, because the points of objection which had been raised, on the one hand, by certain speakers, had been adequately answered, on the other hand, by other speakers. I feel like taking the same position.

However, I should like to correct Dr. Person on at least one point: He referred to the tabular matter dealing with the mechanisms of management, saying it was material that had been used in the Waste Report, and, therefore, lacked value in being represented. As matter of fact, that matter does not appear in the Waste Report, and was not in the possession of the committee that wrote that report. To secure these data, it was necessary to go back to the original questionnaires, about 125 in number, and make a careful study of all of them to secure the information presented in those six tables. It is new information and never has been presented before.

Dr. Person objected to the presence of quoted matter — opinions — but mentioned that the same method was used by the Committee in 1912. The methods used in developing these two reports are identical. The matter presented as quotations, grouped under a number of headings, gives background for the actual experience of men who are engaged in industry, as to the progress which has been made. From those, and other fact material, a number of conclusions have been drawn.

Dr. Person objects further to the failure to present the matter in combination, referring particularly to that absence in the matter of mechanisms. It appears, however, that the matter presented is in combination, for there are twelve items of progress grouped under three heads. Or from another point of view there are twelve items in combination, although only four are explained in detail.

Mr. Jordan brought out the lack of agreement or acceptance of principle. Par. 5 states three so-called regulative principles of management, which were a part of the Report of 1912, and seem to have stood the test of ten years' study and discussion. He also criticized the position of cost accounting, saying it does not deserve to be considered as one of the mechanisms of management. That may be true, but there is a difference between presenting that which you discover, and that which you would like to discover. The position which cost accounting occupies in the six tables is by the results of the 125 questionnaires.

He also regretted that the National Association of Cost Accountants was not included in the list of management societies.

I concur with him in that regret, merely saying, however, that that organization has lived largely within itself, is not well known at the present time in the management field, and that this report was written before the society honored itself by electing Mr. Jordan as its president. I am sure that any consideration of management organizations from this time on must include the National Association of Cost Accountants.

Dean Kimball presented the detached point of view which any report of this kind must seek to take.

Dean Sackett emphasized the educational advance, which unquestionably is one of the most significant and most striking developments of the last ten years. Today some eight or ten of our great national engineering educational institutions have highly organized courses in management, in connection with mechanical, civil, and electrical engineering.

Mr. Rushmore very properly emphasized the need of definitions — of a language of management which we will all understand and use in the same way. A report,[1] one of the first of a series which will attempt to define management terminology and give the common language or grammar which we need, has been presented to the Society.

Mr. Wallace emphasized the points brought out in Pars. 17, 18, and 19 — the necessity of working from the bottom up — and laid particular stress upon economic knowledge. Without doubt, if a study of this kind be made ten years hence, the development in economic knowledge will be found to have been one of the great steps taken in the development of management in industry.

[1] Progress Report of Committee on Standardization of Terminology.

No. 1877

THE DEGASIFICATION OF BOILER FEEDWATER

By J. R. McDermet, Jeannette, Pa.
Member of the Society

The paper deals with the fundamental laws governing the operation of a method of degasifying water fed to boilers and economizers by heating the water to a temperature of some 25 deg. fahr. above the temperature at which it is to be deaërated and then suddenly introducing this water into a chamber in which a vacuum is maintained by a condenser and an air exhauster in series. The successful operation of the process depends upon (a) the inevitable reduction of solubility of dissolved gases in the water with increase of temperature (b) the explosive boiling caused by the rapid injection of heated water into a zone of lower boiling temperature and (c) a method of control and agitation of the water subsequent to the explosive boiling in a reduced air tension. The paper discusses the problem of eliminating air from feedwater in different types of heaters.

DEAËRATION of the water fed to boilers and economizers for the prevention of corrosion is now a commercial realization. Engineering experience has also established empirically the degrees of degasification required, and indicated broadly the field in which it will be useful. One method among several which have met with success has been discussed from these relationships by the author in two previous papers,[1] and he proposes here to deal with the fundamental laws governing the operation of this method, and to indicate the extent of their application to conventional types of feedwater-heating equipment.

2 The method of degasifying water referred to above consists in first heating it to a temperature some 25 deg. fahr. above the temperature at which it is to be deaërated, this latter tempera-

[1] Trans. Am. Soc. M. E., vol. 42, p. 267; and Mechanical Engineering, vol. 43, no. 5, p. 319.

Presented at a meeting of the Baltimore Section of The American Society of Mechanical Engineers, Baltimore, Md., April 12, 1922.

ture being selected by reason of operating conditions. The heated water is then suddenly introduced into a chamber in which a vacuum is maintained by a condenser and an air exhauster in series. The vacuum is so correlated with the heater temperature that the water entering the deaërator chamber is superheated roughly 25 deg. above the temperature of the vacuum. This superheat energy produces a partial flashing into steam, and a pulverization of the liquid as it is suddenly injected. The steam, from the instantaneous boiling, enters the condenser, which is cooled by the supply water on its way to the heater. The heat liberated is recovered by the condenser and recirculated back to the heater. The non-condensable gases originally dissolved are removed from the end of the zone of condensation in the condenser by the air exhauster.

3 The process is significant in that it involves no heat losses. A small quantity of heat, amounting roughly to 25 B.t.u. per lb. of water handled, is continuously recirculated between deaërator and heater, and any degradation of form which it suffers is not significant in heating processes. There are, however, energy charges in removing the water from the region of vacuum and in exhausting the non-condensable gases from the condenser.

4 The successful operation of this process depends upon three factors: (*a*) The inevitable reduction of solubility of dissolved gases in water with increase of temperature. This advantage is common to all forms of water-heating apparatus; (*b*) the explosive boiling caused by the rapid injection of heated water into a zone of lower boiling temperature. While only a very small amount of heat energy is involved, the rate of energy liberation is quite rapid, producing a boiling action, which is independent of diffusion currents within the liquid and results in a very effective disruption of the liquid; (*c*) a method of control and agitation of the water subsequent to the explosive boiling in a region of reduced air tension. This reduction of air tension is secured partly by the reduction of total pressure incident to the vacuum, and partly by the control of the boiling process to furnish a partial vapor component of total pressure to reduce the partial air tension, the sum of the two being equal to the pressure in the region of vacuum. Factors (*b*) and (*c*) in their application are unique with this apparatus, but the control of air tension is significant in any process of aëration or deaëration. This factor of air tension is the criterion by which to judge the performance of other types of

feedwater-heating apparatus from the standpoint of the removal of dissolved gases.

5 The solution of gases from the atmosphere, eliminating carbon dioxide which goes into chemical combination, follows Henry's law. The principal gases in air — oxygen, which produces corrosion, and nitrogen, which furnishes the bulk of the volume — have different solubility constants, and it is expedient, therefore, to consider the application of Henry's law to the individual constituents. Henry's law is formulated by Nernst[1] thus: "Gases dissolve in any selected solvent in the direct ratio of their pressures." When applied to a gaseous mixture this law may be made

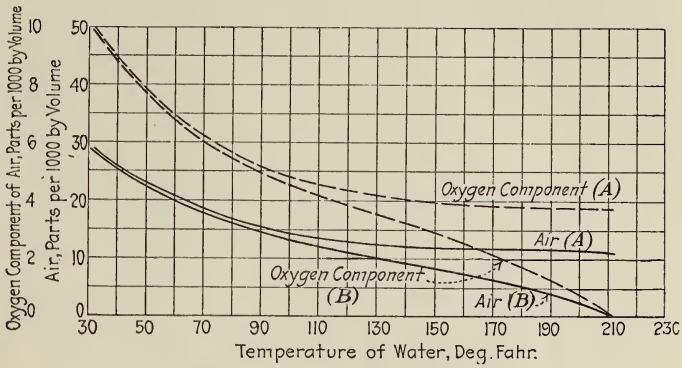

Fig. 1 Solubility Curves for Air in Water

[(*A*) Total pressure of air = 29.92 in. Hg.; (*B*) Partial pressure of air + pressure of water vapor = 29.92 in. Hg.]

more specific by saying that the solubility of any constituent is equal to a proportionality factor, which is different for each permanent gas and itself a function of the temperature multiplied by the partial pressure of the constituent. At any constant temperature the solubility of any constituent for purposes of engineering calculation varies directly as the partial pressure.

6 The solubility curves for air in water presented in Fig. 1 have been recomputed from the solubility data of Winkler. Two sets of curves are given, for both the oxygen component of air and for air, one set having been computed for a total pressure of air equal to 29.92 in. of mercury, and the other so that the sum of the partial pressure of the air and the pressure of the water vapor corresponding to the temperature will be 29.92 in. of mercury.

[1] W. Nernst, Theoretical Chemistry.

Obviously, in the first case the total pressure is indeterminate without calculation, and in the second case the same is true of the partial air pressure. However, the two groups of curves are useful, for between their intercepted ordinates lie the solubility values of air in water for any open feedwater heater operating on raw water and under atmospheric pressure.

7 It is characteristic of an open-type feedwater heater that it operates under atmospheric pressure irrespective of the temperature to which it heats, and that the supplies of water and steam are not interrelated. As a result the control of partial air pressure in the heater depends primarily on the control of venting

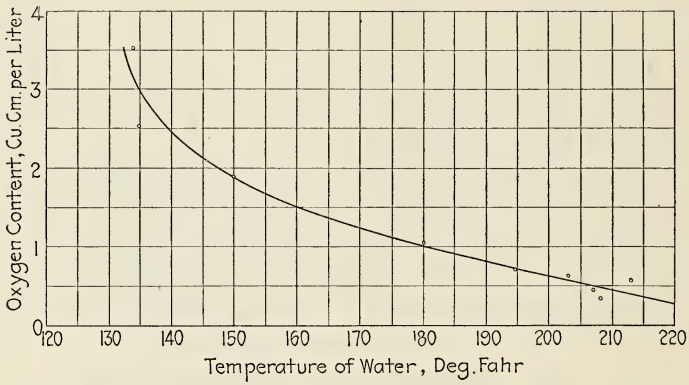

Fig. 2 Dissolved Oxygen in Water in Open-Type Feedwater Heater in Representative Central Stations

(Heaters fed with condensate from condensers.)

if the heater is operated at a temperature close to the atmospheric boiling point. Invariably, there is a high air tension and a very considerable saturation if the heater is operated appreciably below the boiling point. Fig. 2 gives the result obtained from various types of such heaters fed with condensate from surface condensers. The data plotted were collected in a general survey of representative central stations scattered over an area east of the Mississippi River and are indicative of the average performance.

8 One point on this curve, however, is taken from a thoroughfare heater operating at 208 deg. fahr. This heater was installed in a blast-furnace boiler plant, and the entire exhaust of the blowing engine was discharged through the heater. This corresponded to a more extravagant venting than is permissible in standard

practice, but it indicates decisively the results a heater may give under proper reduction in partial air tension. All of the other heaters operate with the minimum amounts of venting consistent with temperature desired and in accordance with usual central-station practice.

9 The results to be expected from any standard heater operating with raw-water feed are illustrated in Fig. 3. One point is significant in this curve — that for an Elliott 1000-hp. open heater — and, in general, is applicable to all types of open heaters. There

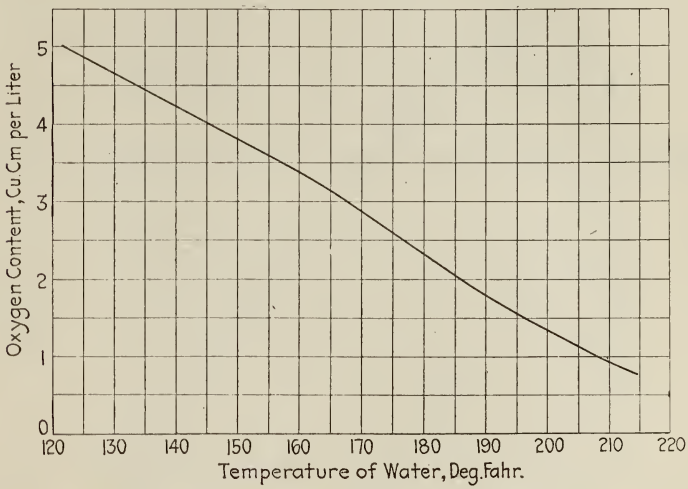

Fig. 3 Dissolved Oxygen in Water in Standard Heaters Operating with Raw-Water Feed

(Water saturated at 70 deg. fahr.)

is a viscosity-surface tension relationship in water which prevents the liberation of air bubbles at temperatures below 160 deg. fahr. As a result, unless considerable precaution is taken, the air-removal results for heaters operating below this temperature are very erratic and sometimes represent no separation at all. As temperatures rise above 160 deg. the solubility values more closely approach the theoretical, but in any event the equilibrium of solubility between gases and water is obtained very slowly, and accordingly, heater results even within this range are sometimes disconcerting. It is possible, however, to calculate, as in Fig. 4, the minimum solubility to be expected from an open heater. This calculation is primarily a proposition in air tension. It is very difficult, if not

impossible, however, to get a heater which will equal this performance on saturated supply water, and the curve, therefore, is only of suggestive significance.

10 The reason for the high values shown in the heater-performance curves is explainable from the curve of Fig. 5, which is plotted for vent mixtures from open feedwater heaters. It is an axiom in condensation work that the most efficient place for the removal of non-condensable gases is at the end of the zone of con-

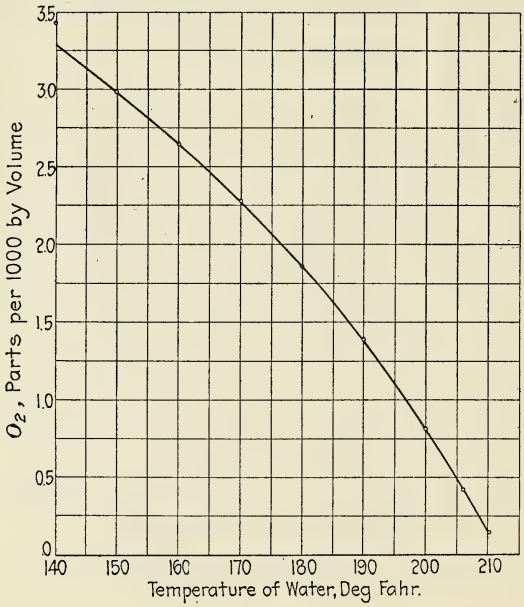

Fig. 4 Minimum Oxygen Saturation to be Expected with an Atmospheric Open Heater

densation. Unfortunately, there is no correlated control of steam and water in an open heater, and as a result no definite zone of condensation exists; it is therefore necessary to consider air removal on a pure vapor-mixture basis. As a result, in order to adequately remove a pound of air, it is necessary to remove a very significant amount of steam, even though the actual quantity of air involved is small. This curve rises so rapidly at 210 deg. as to make adequate venting almost prohibitive from a heat standpoint. It is primarily for this reason that open heaters do not adequately solve the deaëration problem.

11 A series of curves analogous to Fig. 5 but for a condenser

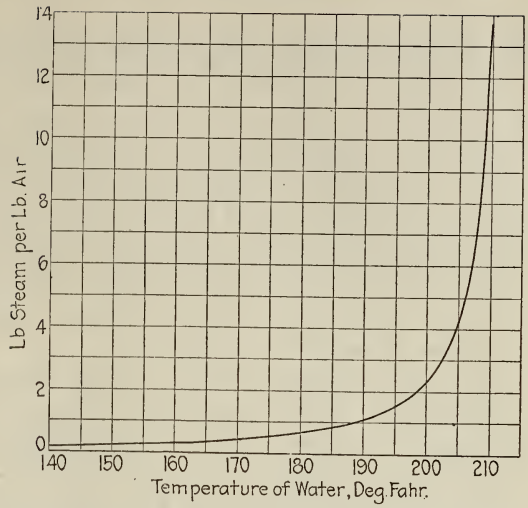

FIG. 5 AIR-STEAM RATIOS IN VENTING ATMOSPHERIC OPEN HEATERS

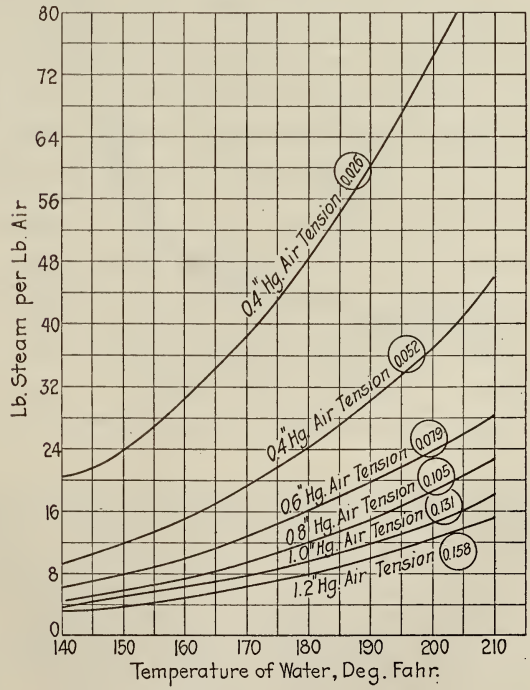

FIG. 6 AIR-STEAM RATIOS IN REMOVING AIR FROM JET-CONDENSER HEATERS

(Figures in circles are mean solubility values in cubic centimeters per liter.)

heater of the jet type, are shown in Fig. 6. Since these heaters operate under vacuum and with different degrees of air exhaustion, the curves are plotted for different air tensions in the condenser body. The condition of control of steam and water exists under the same unfavorable circumstances as in the atmospheric heater, and air-removal conditions are extremely severe. Therefore, the three upper curves, for 0.2, 0.4 and 0.6 in. of mercury air tension, are of comparatively little significance, and the expense of pumping out the air mixtures has practically relegated the zone of operation to air tensions within the zone of the three lower curves, for 0.8, 1.0 and 1.2 in. of mercury air tension.

12 In the open feedwater heater, venting is secured at the expense of steam only. In the jet-condenser heater, operating at vacuum, the vapor mixture must be mechanically exhausted. From a steam-ejector standpoint it requires practically as much energy to evacuate a pound of steam as it does a pound of air. Jet-condenser heaters have uniformly been more efficient aërators than deaërators, although their other merits are making their use extremely popular. In one installation, which perhaps may be said to represent the best from the standpoint of deaëration, the solubility of the water leaving the condenser is practically regulated by the condensate coming over from the main turbine condenser. In this case the jet condenser removes no air, but fortunately does not allow any to be added. It is also true that equilibrium between air and the solution of air in water is so slowly attained that spraying methods are uniformly unsuccessful in producing complete deaëration. There is, therefore, little probability that the jet-condenser heater, even with extravagant air-removal capacity, will offer a successful solution. It has, however, under the best conditions of heating, marked one step in advance, in that it has been found capable of preventing pollution of the water handled.

13 The use of surface-condensing apparatus has two inherent disadvantages. While the condensate or water which is to be heated does not come in contact with air, the use of such apparatus for exhaust steam from small turbines under modern boiler conditions is limited by the conductivity of superheated steam. The exhaust from small auxiliaries is so high in superheat under high boiler pressure and high boiler superheat that it is almost impossible to secure a workable conductivity without desuperheating. Where desuperheaters are employed or steam is extracted

from an intermediate stage of the main unit within the saturated zone, the problem is complicated by the aëration of the condensate in the heater.

14 Fig. 7 indicates the analogous condition of control of air tension in the deaëration process previously described. The scale of ordinates for these curves is inches of mercury air tension, and the scale of abscissas the B.t.u. drop in the instantaneous boiling. The various curves range between the maximum and minimum

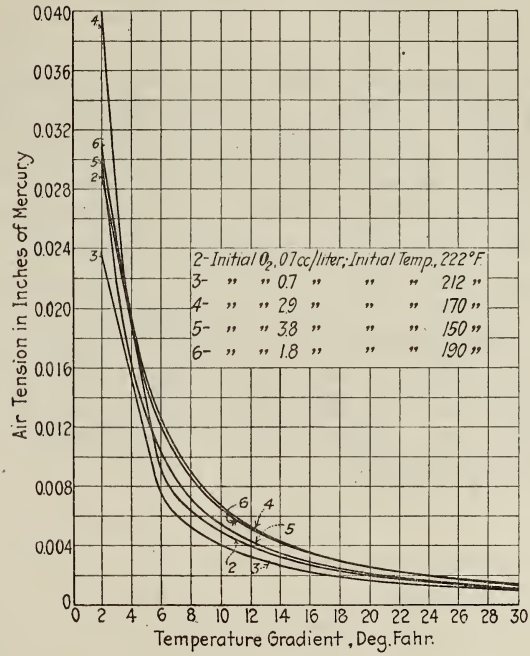

Fig. 7 Characteristic Curves of Deaërator Described in Text

operating temperatures employed in the process and for initial air contents which ordinarily accompany these temperatures in feed-water heaters of the open type. Obviously, the closed type merely retains the air content of the water entering it. It is a fortuitous circumstance that the natural coördination between normal air contents in heating apparatus and vacuum are such that these curves even for widely different conditions are approximately parallel and lie within the same zone. The minimum air tensions in the operating range amounting to less than 0.002 in. of mercury, indicate very forcibly the reason for the effective results which are

secured by this process. However, the results which are theoretically obtainable are decidedly lower than the results actually obtained. The water is handled in the region of vacuum by agitating pans, which as a general premise are very much more effective than any type of spray nozzle. These curves are not in any sense an exposition of the completed operation of the deaëration process under consideration, but they do explain very satisfactorily and very accurately the underlying principles upon which successful operation is based, and indicate in radically new ways the relation of these principles to other features of power-plant apparatus.

15 Some mention was made in an earlier paragraph of the fact that a gradient of 25 deg. was employed between the heater and the deaërator. For all conditions of pressure the characteristic curves become practically parallel to the axis of abscissas in the neighborhood of 25 deg. There is no appreciable gain from increasing this value and it is economically wrong to extend the range of temperature gradient unless it be for some purpose of regulation as a part of a complete power plant. It is true, however, that the process does not operate efficiently at heater temperatures below 160 deg. fahr., and in case it is desired to go to operating temperatures on the deaërator as low as 130 deg. fahr., the temperature range must be extended beyond 25 deg., regardless of economic proportioning.

No. 1878

LUMBER DRY KILNS

By Thomas D. Perry, Grand Rapids, Mich.
Associate-Member of the Society

The three main divisions into which lumber dry kilns may be grouped are blower, condenser, and ventilated kilns. The author, who believes that scientific kiln drying offers engineers a splendid field for research work, describes these types and discusses the possibilities of each, with particular reference to the ventilated kiln, of which there are several classes. The questions of moisture deficit and equilibrium, air interchanges, and drying cycles are adequately treated, and data showing the relation between relative and absolute humidity during the drying period are included.

THE PROBLEM presented to the manufacturer who would utilize woods intelligently is a diversified one owing to the fact that the score or more of major varieties of commercial timbers are grown in climates and seasons differing widely in temperature and moisture, and on soils ranging from the rocky hills of the Appalachians to the rich alluvial lowlands of the Mississippi delta. The origin of the lumber has a noticeable effect on its water content. Lumber or veneer (thin lumber produced usually by rotary cutting or flat slicing, sometimes by sawing), when produced from the log, contains a large proportion of water, ranging from 25 to 75 per cent of the total weight. One square foot (board measure, one inch thick) of gum lumber, weighing approximately five pounds when sawed, will be reduced to about three pounds when its water content of approximately one quart has been evaporated. Oak grown on a hillside may contain only a pint (approximately 1 lb.) and swamp gum may have from two to four pints of water per sq. ft. B. M.

2 This water content of wood exists in two forms, free moisture and cell moisture, the former being readily evaporable in ordinary air drying, and the latter demanding excessive air drying

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(several years) or artificial treatment in kilns. The usual border line between the two forms of moisture is in the vicinity of 30 per cent moisture content (the percentage of the weight of water removed being computed on the dry fiber weight as a base). It is possible to use artificial means to remove this free moisture, but a simple air exposure is usually more economical.

3 By far the largest volume of lumber products are dried in the form of lumber or veneer. The drying of unusual forms and shapes, such as staves, handles, shingles, laths, etc., is decidedly specialized and outside the range of this discussion.

TYPES OF KILNS

4 The original artificial drier was a smoke kiln, now practically obsolete. This was followed by the furnace kiln and the steam-coil kiln. Some woodworkers still persist in using home-

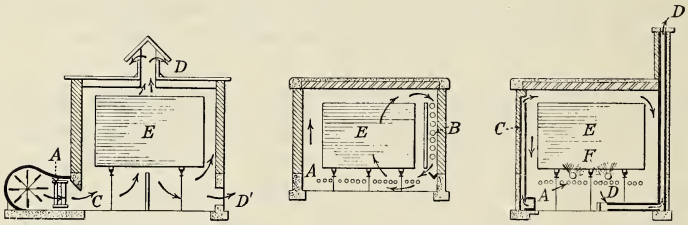

Fig. 1 Showing (from Left to Right) Blower, Condenser and Ventilated Kilns

(A, Heating unit; B, condensing unit; C, fresh-air inlet; D and D', used-air outlets; E, lumber charge; and F, humidifying unit.)

made equipment, but kiln design and building has become a special branch of manufacturing that has its own recognized field.

5 The three main divisions into which lumber kilns may be grouped, as illustrated in Fig. 1, are:

1 Blower: Mechanically forced ventilation, or recirculation, whether suction or plenum method; moisture-laden air usually discharged out of doors

2 Condenser: Generally of the gravity recirculating type, in which the air passes over moisture-removal or condensing units, once in each interchange

3 Ventilated: Fresh air taken in and used air discharged direct from kiln to atmosphere, utilizing the fundamental laws of physics to obtain internal circulation.

6 To understand the particular range of each of the foregoing types it is necessary to outline the moisture-removal problem in some detail from the standpoint of the lumber to be dried. Air drying removes more moisture from the surface than from the center, and owing to the length and width of a board, drying takes place chiefly through the flat faces, rather than through the ends or edges. The inevitable result is a surface drier than the interior, and air-dried stock is therefore subject to an internal strain that often manifests itself in the form of warp, twist, or other surface irregularities. The problem is to draw the center moisture out and have surface and interior equally dry.

7 The skill of an engineer is not required to discover that, if the drying of wood afforded no organic difficulties, the blower would dry rapidly, utilizing considerable power, and the condenser would function slowly but at low cost.

8 The ventilated kiln is the least understood, even though most generally used, and offers an unusual opportunity for speed and efficiency when its underlying principles are grasped.

9 The nature of wood is the phase of the problem that engineers least appreciate. It is almost axiomatic that wood should not be subject to external or internal strain during drying, but it is practically impossible to obtain such a "strainless" condition. The reduction of this drying strain to a minimum point is necessary to drying without damage to the lumber, the usual manifestations of which are checking, warping, honeycombing, "hollowhorning," etc. The chance of internal strain greatly increases with the thickness of the wood to be dried: e.g., $\frac{1}{16}$-in. veneer is practically all surface and can develop little internal strain in any kind of drying, while 4-in. green oak presents a decidedly stubborn drying problem, and taxes the skill of the best operator in an efficiently controlled kiln.

BLOWERS AND CONDENSERS

10 These facts lead to the logical conclusion that where thickness is nominal, as in the case of veneer, applied shellac, stain, filler, varnish, and glue, the speed of the air movement in a blower kiln will remove moisture rapidly and cause no serious damage; but where thickness becomes appreciable ($\frac{1}{2}$ in. and up) the rapidly moving atmosphere of a blower kiln will produce uneven drying and unnecessary interstrain that will inevitably damage the lumber. Even with attempted maintenance of high humidities (difficult in

blower kilns [1]) the hazard is serious. On the other hand, the slowness of the condensing kiln will make for accurate control, but the lack of speed may force the initial cost of installation to an excessive amount. As a matter of fact, Government experts have frequently recommended the condensing kilns without comprehending in most cases the economic aspect. It has been the author's experience during the last five years that kilns of the condensing type used for war work did no better drying than the best of the ventilated types, and required for an equivalent output practically twice as many units, at approximately double the initial cost per unit, or a quadruple investment, with no appreciable gain in quality or safety.

11 As a conclusion it may be noted that blower kilns are most suitable where thickness is not a factor, and that condenser kilns are especially serviceable in the drying of thick green woods where internal strain is a decided danger.

TYPICAL DRYING CYCLE

12 Before considering the individual characteristics of the various types of ventilated kilns, it may be well to outline briefly a drying cycle. Take an actual operation schedule, tested by practice, of reducing 1-in. oak from 35 to 5 per cent moisture content in 16 days of 24 hours steam. Plotting this schedule, curves showing the variations in temperature and the relative humidity during the period of drying are obtained as shown in Fig. 2. The drying cycle divides itself into three phases: initial "steaming" or high-humidity period in which lumber is heated through and enough moisture added to make the surface as wet as the center; intermediate "stewing" or cooking, a transition period ("stewing" used in the sense of a vapor bath, rather than as cooking [212 deg. fahr.]. Some authorities merge this into the "drying" period); and the final "drying," when humidity may be dropped and temperature raised within limits demanded by the kind and condition of lumber.

13 The temperature should gradually rise from 125 to 170 deg. fahr. and care should be exercised that it does not go up too fast during the initial steaming, which would tend to crust the outside of the boards. The relative-humidity line reaches 100 per cent as rapidly as possible without producing an excess of temperature. The reason for the high initial humidity is that partly dried lumber,

[1] Kent's Mechanical Engineer's Pocket Book, 1916 edition, p. 573.

as placed in the kiln for drying, normally has a surface drier than the interior, and unless the surface is thoroughly moistened the internal moisture will be sealed in by the dry and shrunken surface layers; in other words, the degree of internal strain will be intensified rather than reduced. If the surface of the board is steamed or exposed to an atmosphere of high humidity it will absorb moisture and expand, making it possible for the cellular structure to conduct additional moisture to the surface, and reducing the internal strain. Heating the lumber through is also accomplished quickly and safely by this steaming.

14 Moisture will pass out more rapidly from the surface of a

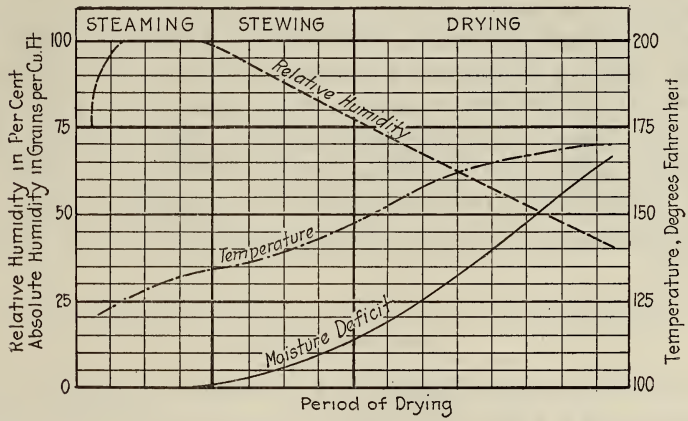

Fig. 2 Temperature and Relative Humidity During Drying Cycle and Curve Showing Moisture Deficit

board the center and surface of which are approximately equal in moisture content, and the final result of the drying cycle will be a reasonably uniform moisture content from face to face of the board and an absence of internal strain, with consequent damage. It is a rather significant fact that lumber dried by the so-called " wet " process explained in the preceding paragraph is more plump than when treated by the old-fashioned method of " giving it all the heat it will stand " without preliminary steaming or due regard to the humidity. Experiments have proved that the endeavor to dry without steaming takes longer because moisture transfuses more slowly through a dry than a wet surface, and the resulting damage is usually serious.

RELATIVE VS. ABSOLUTE HUMIDITY

15 Another way of expressing the function of humidity is to express the cycle of drying from the standpoint of absolute humidity, or the grains of water vapor per cubic foot of atmospheric kiln content. Table 1 shows the relation of absolute (grains of water per cubic foot) and relative humidity (percentage of saturation) within the usual ranges of kiln temperature.

16 If we could isolate and examine a cubic foot of space (containing air and water vapor) in a kiln at a temperature of 150 deg. fahr. and a relative humidity of 50 per cent, we would find that it contains 36.76 grains of water in the form of vapor and possesses the capacity, before reaching saturation, of absorbing an equal additional amount of water vapor. It is this difference between actual grainage and the grainage of saturated water-vapor content that expresses the drying power of the kiln. It is obvious that the higher the temperature, with a given moisture deficit, the more rapid the evaporation. Stated in another way, this means that a cubic foot partly saturated has a tendency to become entirely saturated if free moisture is accessible, i.e., from the lumber.

MOISTURE DEFICIT

17 This moisture deficit may be termed the measure of drying power in a kiln (when considered in connection with the temperature and circulation) and the greater the deficit the greater will be the pull exerted on the moisture contained in the lumber. At the beginning of the drying operation, when reducing the lumber to a uniform wetness, the moisture deficit is nil; and toward the end of the operation, at a temperature of 170 deg. fahr. and a relative humidity of 40 per cent, each cubic foot is eagerly seeking for 67 grains of water vapor or such part of it as can be extracted from the lumber. Fig. 2 contains a curve showing the moisture deficit expressed in grains per cubic foot. It will be noted that this line curves up more rapidly than the temperature line. This moisture deficit may be termed the measure of potential drying power. It is easy to remove moisture from the lumber at first, but the last few stages of drying require a decidedly strong pull.

18 While it would be possible to reduce the relative humidity to below 40 per cent (carrying the moisture deficit above 67 grains),

it would result in too rapid a rate of surface drying and leave an unevenly dried product. The secret of controlling temperature and humidity in efficient drying is to avoid any sudden changes during the progress of the operation and to finish without carrying the temperature to an extreme that will damage the wood; and to avoid reducing the relative humidity at any point where it will dry the surface faster than the center.

AIR INTERCHANGES

19 The number of air interchanges per hour should not be too great. If an anamometer test shows three or four complete air changes per hour, it is enough. The process of the withdrawal of water vapor from the interstices of the lumber cannot be very rapid. Too rapid air movement in a gravity kiln subjects it to the same criticism as a blower kiln, i.e., too rapid surface drying.

20 High humidity and large circulation of air are antithetical to one another. To obtain high humidity, the circulation must be either stopped altogether or greatly reduced, and to reduce the humidity a greater circulation must be induced by increasing the draft.

21 The following example, based on three interchanges per hour, illustrates this point:

Size of kiln inside	15 ft. high by 19 ft. wide by 26 ft. long
Cubic contents, gross	7410 cu. ft.
Lumber allowance	1083 cu. ft. (13,000 ft. B. M.)
Equipment allowance	50 cu. ft.
Cubic contents, net	6277 cu. ft.

Allowing a moisture-lifting capacity averaging 25 grains of water vapor per cu. ft. of each discharge from the kiln:

Discharge per cu. ft.	25 grains or 0.00357 lb. avoirdupois
Total single discharge	$0.00357 \times 6277 = 22.42$ lb.
Hourly discharge (3)	67.26 lb.
Daily discharge (72)	1614.25 lb.

Estimating the average water content of saturated oak lumber at 1 lb. per ft. B. M.:

Lumber allowance	13,000 ft. B. M.
Moisture to be removed	13,000 lb.
Time required	$13{,}000 \div 1614 = 8.05$ days

22 This would indicate that the actual drying period (Fig. 2) will require practically eight days if the texture of the lumber permits of an average removal of 25 grains. An additional time allowance should be made for steaming.

TABLE 1 GRAINS OF WATER VAPOR PER CUBIC FOOT AT VARIOUS RELATIVE HUMIDITIES[1]

(7000 grains = 1 lb. avoirdupois)

Temperature, Deg. Fahr.	Relative Humidity																			
	5%	10%	15%	20%	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%	80%	85%	90%	95%	100%
100	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
105	1	2	3	5	6	7	8	9	10	11	13	14	15	16	17	18	19	21	22	23
110	1	3	4	5	7	8	9	10	12	13	14	16	17	18	20	21	22	24	25	26
115	2	3	5	6	8	9	11	12	14	15	17	18	20	21	23	24	26	27	29	30
120	2	3	5	7	9	10	12	14	15	17	19	20	22	24	26	27	29	31	32	34
125	2	4	6	8	10	12	14	16	18	20	22	23	25	27	29	31	32	35	37	39
130	2	4	7	9	11	13	15	18	20	22	24	26	29	31	33	35	37	40	42	44
135	3	5	8	10	13	15	18	20	23	25	28	30	33	35	38	40	43	45	48	50
140	3	6	8	11	14	17	20	23	25	28	31	34	37	39	42	45	48	51	54	56
145	3	6	10	13	16	19	22	26	29	32	35	38	42	45	48	51	54	58	61	64
150	4	7	11	14	18	21	25	29	32	36	39	43	46	50	54	57	61	64	68	72
155	4	8	12	16	20	24	28	32	36	40	44	48	52	56	60	65	69	73	77	81
160	4	9	13	18	22	27	31	36	40	45	49	54	58	63	67	72	76	81	85	90
165	5	10	15	20	25	30	35	40	45	50	55	60	65	71	76	81	86	91	96	101
170	6	11	17	22	28	34	39	45	50	56	61	67	73	78	84	89	95	101	106	112
175	6	12	19	25	31	37	44	50	56	62	69	75	81	87	94	100	106	113	119	125
180	7	14	21	28	34	41	48	55	62	70	76	83	90	97	103	110	117	124	131	138
185	8	15	23	31	38	46	54	61	69	77	84	92	100	107	115	123	130	138	146	153
190	8	17	25	34	42	51	59	68	76	84	93	101	110	118	127	135	144	152	160	169
195	9	19	28	37	47	56	65	75	84	94	103	112	122	131	140	150	159	168	178	187
200	10	21	31	41	51	62	72	82	92	103	113	123	133	144	154	164	175	185	195	205

[1] Adapted from tables compiled by Dr. William M. Grosvenor, with intermediate readings interpolated and decimals discarded.

23 The removal of 2 lb. moisture per ft. from green gum, for instance, would necessitate either increasing the drying time proportionately, or operating at a higher temperature.

MOISTURE EQUILIBRIUM

24 Theoretically it may be possible to reduce lumber to absolute dryness, but it cannot be kept so except in sealed receptacles. Under ordinary factory-workroom conditions, it will reabsorb to 6 or 8 per cent moisture. It is necessary, therefore, only to dry down below normal and allow for reabsorption. In ordinary lumber yards the lumber will rarely air-dry below 15 per cent, and kiln-dried stock left out of doors will absorb to about the same point. For structural purposes, exposed to weather the moisture

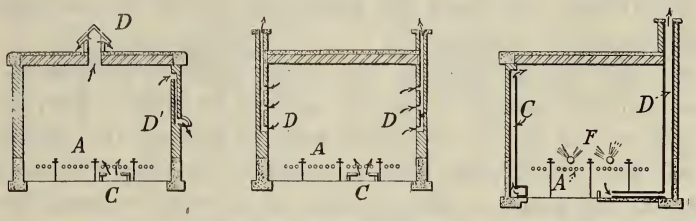

FIG. 3 CLASSES OF VENTILATED KILNS

(From left to right, ventilated at top, side and bottom. *A*, heating unit; *C*, fresh-air inlet; *D* and *D'*, used-air outlets; and *F*, humidifying unit.)

content of timbers should be around 15 per cent to prevent shrinkage or expansion. For more accurate work the equilibrium point should be ascertained for each plant and will be found for interior trim to be about 8 to 10 per cent and for furniture, pianos, etc., from 4 to 7 per cent.

VENTILATED KILNS

25 Ventilated kilns are readily grouped into three classes as indicated in Fig. 3.

26 Kilns of the first class are ventilated at the *top,* either through the ceiling and a cupola on the roof, or through openings in the side walls near the ceiling. Fresh air is usually admitted near the floor and the air movement must be distinctly upward. The real problem is to obtain an efficient utilization of this air movement to lift moisture out and up from the lumber.

27 No difficulty will be encountered in securing ample air discharge, as hot air will always rise and seek an exit. It is obvious that the hot air escaping will not be heavily charged with moisture. Only to a limited extent is it possible to use this upward-moving air as a moisture vehicle, because the increased water-absorption capacity accompanying higher temperatures will carry only a small amount of water vapor without a net increase in weight.

28 As an example, suppose the temperature of the air in the lower part of the kiln is 140 deg. fahr. and at the top 160 deg. fahr.; dry-air weights, 463.09 and 448.11 grains per cu. ft., respectively, a difference of 14.498 grains (less than 20 per cent relative humidity at 160 deg.). Air, therefore, cannot move upward and carry as much as 15 grains (neglecting expansion which is less than 4 per cent). Compare this with the fact that in a down-draft kiln the air at 140 deg. can carry 56 grains per cu. ft. before reaching saturation.

29 Any steam spray in this type tends to monopolize the limited moisture deficit of the kiln air and to greatly decrease the drying power as the air circulates upward through the lumber.

30 The second class have air exits in the *sides* and air inlets usually at the bottom. There is less of a scientific basis for this type than for either the first or the third type. There is great danger of "short-circuiting" the air across a corner of the kiln, thus causing pockets in places where a lack of air movement prevents drying. When this type is applied to the primary drying (to reduce shipping weight) at sawmills, with high temperatures, it may give fair results in regard to the removal of free moisture, and perhaps will reduce the moisture content to 15 per cent, but below this point it is undependable.

31 The use of steam spray or the handling or relatively moisture-laden atmosphere is no more possible than in the first type. Neither of these types is adapted to what might be termed final drying for musical instruments, furniture, high-grade cabinet work, interior trim, or any object where permanence of dimension and a high degree of finish are required.

32 The third type, which is often called "reversed" ventilation, has a downward circulation with outlets at the *bottom*. The fresh-air inlets are at or near the ceiling line and the damp-air exits are in the pit below the lumber. This method adapts itself most effectively to the handling of heavy moisture-laden air shown

to be necessary to dry the lumber uniformly. Whenever any portion of the atmosphere of the kiln acquires a substantial amount of moisture, it will lose temperature, increase in weight, and settle. It is therefore necessary to provide an adequate damp-air accumulation pit below the lumber, and to supply chimneys or stacks with sufficient draft to carry this moisture-laden air up and out. Heavy air that has settled into a pit will not readily be drawn up a stack. Steam-heated pipes in the stacks will accomplish this and at the same time keep the water vapor from condensing on the interior of stacks or at their top outlets. This type gives opportunity for excellent control, as any humidity can be handled and the air discharge be under positive control under various climatic con-

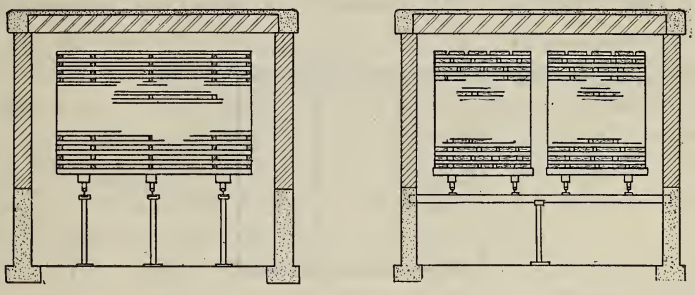

Fig. 4 Transverse Section of a Kiln, Showing Cross-Pile (at Left) and End-Pile (at Right)

(Notice somewhat better air circulation opportunity in end-pile.)

ditions. For high-grade work it affords reliable regulation of the three essentials: temperature, humidity, and circulation.

33 It has been the purpose of this paper to outline some of the fundamental principles in lumber drying and to indicate their most important applications. Many of the combinations of mechanical apparatus are adequately covered by United States and foreign patents, although the fundamental elements have been in use too long to have further protection.

EXISTING STANDARDS

34 A few standards of construction, design, and operating methods are included to assist in general layouts and development sketches.

35 Kilns may be *end-pile* or *cross-pile,* i.e., boards may be parallel or at right angles to the rails (Fig. 4), depending largely

on the factory and yard arrangements. End-pile gives slightly better internal circulation, although this is rarely determinative.

36. A kiln car of lumber is usually 6 ft wide, 9 ft. high (above rails including trucks), and from 12 to 16 ft. long (rarely longer). Higher loads not only upset easily but greater heights introduce uneven drying, or wet pockets, because of unavoidable temperature differences between the bottom and top of an enclosed space. The writer knows of a battery of kilns 20 ft. high in which the top half of the load always dried first and there was a temptation to discharge the kiln before the bottom half dried.

37 Kilns may be of the *progressive* (open at both ends) or *single charge* (usually open one end) type as shown in Fig. 5. The former involves a daily opening to remove and enter lumber,

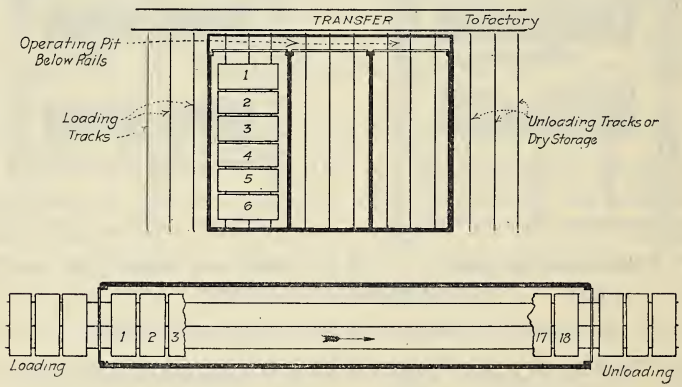

Fig. 5 Progressive Kiln (Below) and Single Charge (Above)

(Each has equal holding capacity and output, together with minimum necessary trackage for loading and unloading.)

a daily moving forward of every car, with a consequent loss of from 5 to 10 per cent of the drying period from the opening of kiln, during the time required to cool sufficiently to move the cars, to the time when temperature and humidity are restored to operative conditions. The progressive kiln, to be efficient, requires that the charging end be maintained at a low temperature and a high humidity and the discharging end at a high temperature and low humidity with graduated intermediate conditions. Air circulation, chiefly transverse, prevents dependable graduated conditions and the operator copes with irregular conditions. The subdivision of the heat and humidity sources may be of slight

help but in any case the progressive kiln is more complicated than the single-charge type.

38 The latter type (sometimes called box, compartment, pocket, or cell) requires the operator to vary his entire kiln regulation from day to day according to a predetermined schedule. The room is small enough so that control of uniform conditions throughout is simple. Various kinds and thicknesses of lumber can be separated in different rooms and given individual treatment. Valves and dampers can be located in an accessible outside pit. When built in batteries with adequate transfer and yard trackage the single-charge type will produce as great a turnover from the investment as the progressive, and dry the lumber better.

39 Hollow tile, with an exterior brick veneer, makes the best walls, and a tile and reinforced-concrete roof with insulated wooden doors will complete an economical heat-retaining building. Solid concrete is not only a poor insulator but also absorbs too much moisture and checks badly under high temperatures. Frame buildings, under the varying conditions of temperature and humidity, rot and rack rapidly.

40 The output of many kilns is seriously hampered by lack of foresight in providing adequate loading tracks, transfer arrangements, dry storage rooms and unloading facilities. Kilns will produce their full capacity only when there are no delays in charging and discharging.

41 Although this paper is too limited to include descriptions of detailed operating schedules, recording and regulating instruments, or methods of testing and tabulating results, it may be stated that any type of kiln gives its best results only to the intelligent operator who is methodical and careful in his adjustments, and who keeps such comprehensive records as to prevent making the same mistake twice.

42 Scientific kiln drying is only at its beginning and wood users have but commenced to visualize the efficiency and economy that is possible. Engineers can do much toward the solution of the problem by giving it the same degree of study and research that has resulted in substantial progress in other fields.

DISCUSSION

C. A. Cutler. The author dismisses blower kilns for lumber drying with the few trite words that they "will produce uneven drying and unnecessary interstrains that will inevitably damage the lumber." It would be unfair to the person wishing information on lumber kilns to let such a statement go unchallenged. He gives as his reason for this statement "rapidly moving atmosphere" and "difficulty in maintaining high humidities."

It is impossible to dry lumber at an even rate or in the least time unless the speed of air circulation in the kiln is rapid enough to bring each cubic foot of air in the piles back to the conditioning point before it has lost enough of its heat energy or increased its relative humidity to the point where its drying capacity is appreciably less than when it started on its circuit of drying, and nothing but rapid circulation will accomplish this. The prevention of "interstrain" is simply a matter of conditioning the air entering the lumber piles.

Assume a cubic foot of air at a temperature of 160 deg. fahr. to be in motion up through or across a pile of wet lumber. It will travel at least six feet up and down or across the lumber pile. If it does the work it is supposed to do it will reduce in temperature and increase in humidity during every foot it travels. Its ability to absorb moisture is controlled by its temperature and relative humidity. Each board it passes will reduce its drying capacity so that the last board it passes will lose less moisture than the first. The first board it passes will be that nearest in circuit with the heating means and naturally will dry first; the last board to dry will be at the point where the air came out of the pile. The faster the speed of the air the less difference there will be between outgoing and ingoing air and more even will the drying be. Speed of circulation, then, is the governing factor of even drying.

If the circulation is sluggish and slow as in gravity circulation kilns, it will start to rise in the line of least resistance until the upper part of the compartment has been filled with air of the same or nearly the same temperature. Then it will gradually seep into the spaces between the lumber. As soon as it becomes cooler, by reason of having taken on some moisture, it will slowly fall until it gets into a level of equal temperature or less humidity where it will remain until by gradual change of kiln condition it reaches the outlets at the bottom as in the case of Mr. Perry's recommended type of kiln. Hot

air going outside the piles and cold air coming down inside surely will not dry evenly.

In gravity circulating kilns the air is being heated by the coils directly under the lumber (the bottom layers of which lumber have been subjected to direct radiation) and starts in its upward course more or less rapidly, depending upon the temperature of the air and the coils. A slight amount of steam admitted to a manifold for slight adjustment of kiln temperature will heat most of the outgoing pipe for a short distance only, so that often there will be a rising current at and near the steam inlet and a downward current a few feet farther away, but assuming this did not occur and that a rising current of hot air for a few feet and downward current of a few feet of colder air would dry a sixteen-foot board full length evenly, let us go back and follow the "assumed given foot of air" under the conditions that exist in a ventilated type kiln.

It must be obvious that the air will follow the lines of least resistance and that it will go up along the outsides of the piles before it will go up through the piles. There is no way of its getting into the center of the pile until the air in the center of the pile has fallen down out of the pile, because it is colder than the air under it. This tendency to go up outside of the pile and to come down inside the pile must result in drying the outside and top boards long before the center boards.

Now let us assume a blower-type kiln where the heated conditioned air is moved crosswise of the kiln through not more than six feet of lumber. Assume that it is fed to the ingoing side at a speed that will create a slight pressure and that on the opposite side there is a slight negative pressure caused by the suction of the blower. Assume further that these conditions are sufficient to cause all the air between layers of lumber to be removed and replaced at least fourteen times per minute. If one had control of the condition of the air entering the pile and knew just what were the best drying conditions, then surely one could maintain those conditions all through the pile and all the lumber would dry evenly all through the pile.

High temperatures and high relative humidities will reduce moisture contents of lumber by vaporization of moisture without creating any surface strains, but these two conditions cannot be created evenly through piles of lumber without rapid circulation, and rapid circulation cannot be created by anything less than a blower. On the other hand, all practical lumber men know that lumber will air dry more during the cold month of March than during the

hot summer months. The blustering cold winds of March cause a pressure on one side of the pile in the yard, thus blowing out the nearly saturated air that otherwise has to find a slow and roundabout way out by difference in gravity. Speed of air between layers causes this speed of drying — not heat. Then it follows that even a cold rapid circulation will dry faster than a much higher temperature without circulation.

Lumber drying in piles is proportionately wettest at center of piles unless a circulation of air is established and the center of the pile approaches the same dryness of the outside as the speed of circulation is increased, up to the point where the speed of surface drying is faster than the speed of feed of moisture through the cells of the lumber. This speed of feed of moisture is affected by the temperature of the lumber and by its dryness. A dry sponge will not take water until it is wetted. Dry wood cells do not rapidly absorb moisture from wet ones, yet the moisture from the wet center cells must be dried out by absorption of the moisture by the dryer outside cells. Therefore, care must be taken not to dry the outside cells beyond the points where they will act; but it is very simple in a rapid circulation kiln to obtain and maintain a relative humidity that will keep the cells at a degree of dryness only a little less than the center cells. In a blower kiln where the air travels at high speed between each layer of boards for a distance of only six feet and then comes out of the pile back to the conditioning point, it is not difficult to raise or lower either the temperature or humidity to the desired point before it is again circulated. There is another advantage in rapid circulation in that if properly directed, it insures even drying condition in every part of the kilns and piles because if there was by reason of poor construction or piping an uneven distribution of heat it could be mixed with other air before it had time to affect the drying rate.

Mr. Perry mentions an "actual operation schedule" tested by practice of reducing one-inch oak from 35 to 5 per cent moisture content in 16 days. A properly constructed kiln of the blower type can do this same feat in eight days and this difference in drying time is due to being able to maintain ideal drying conditions all the time in all parts of the piles. Ideal drying conditions not only reduce the drying time, but also as might be expected, produce quality drying, practically no internal strains and very even moisture distribution.

The idea of steaming lumber in the initial drying stages, of course, does tend to even up any difference in moisture contents between the

outside and inside of the lumber, but it has a very much greater value of heating the lumber to the kiln temperature. Except under conditions of 100 per cent relative humidity, the temperature of the lumber is always lower than the kiln temperature in proportion to the speed of drying. The employment of high humidities occasionally during the drying will speed up the drying time, but only in rapidly circulating kilns can this be done evenly through the piles. Within the ordinary range of kiln temperatures no damage may be done if the humidity is maintained at or near 100 per cent, but damage may be done when the humidity is reduced too rapidly. Again, it becomes obvious that control of humidity is very important, but how can it be accurately controlled if the circulation is so slow that widely different conditions must be created in one part of the pile of lumber before the air in another part will move out. In a blower kiln, with a speed of fourteen changes of air per minute between layers of lumber, the humidity and temperature will not have a greater difference than 5 per cent and 5 degrees respectively at any time.

Mr. Perry says that "three or four complete changes per hour are enough." If he means that all the air in his kiln must be taken out and replaced by fresh air, it might be all right, but he goes on to say that "too rapid air movement in gravity kilns subjects it to the same criticism as a blower kiln, i.e. too rapid surface drying." The assumption that blower kilns create "too rapid surface drying" is unfair. It must be perfectly obvious that the drying rate is determined by temperature and relative humidity and that these two, so far as the conditions inside the piles are concerned, are controlled by the speed of movement of the conditioned air. Therefore, the rate of surface drying is under better control in a blower kiln than in a gravity circulating kiln.

In contrast with what Mr. Perry says is necessary to obtain high humidities in a gravity kiln, consider what can be done in a blower kiln. In the gravity kiln "circulation must be stopped or greatly reduced" so that moisture can pile up in the kiln. In a blower kiln a valve is opened into the blower duct and within ten or fifteen minutes all the air in the kiln and piles will be at saturation point. The moist warm air is immediately and evenly applied to every board in the kiln. There is no waiting or slowing up of circulation or uneven application of the humidity, all due to speed or circulation. It always comes back to a matter of circulation. The heat energy is supplied at a certain point, yet it must be distributed evenly all over the kiln and through the piles. The more rapid the

circulation, the more even will be the distribution. If the air is forced back to the heater with only a loss of 5 degrees in temperature and is thoroughly mixed with all other incoming air before delivery to the heater, then heated a matter of 5 degrees again it wouldn't be difficult to imagine a more perfect drying circuit, and the essential point of it all is the rapid circulation.

The logical conclusion is that the best and fastest lumber drying ought to occur in kilns with rapid forced circulation.

Burritt A. Parks. While the writer can add nothing to the discussion in the way of offering additional information on the subject, he would like very much to have the discussion bring out more explicitly the relative advantages of the blower type and ventilated type kilns.

Among the advantages claimed for the blower type kiln are the following:

1 If the kiln is properly loaded, using end piled trucks, a more uniform circulation of the air is obtained across both surfaces of the lumber, resulting in more rapid and uniform drying

2 With practically all the air being recirculated the temperature and humidity throughout the kiln are more uniform and more readily controlled

3 Rapidity of circulation under absolute control, where steam engine driven fan is used, making it possible to control kiln conditions in accordance with stock to be dried.

Some of the ventilated and condenser type kiln advocates claim great advantages for their kilns over the blower type on account of no power being required. This has always appealed to me as "sales talk" as the exhaust steam from a steam engine, used to drive the fan, can be used in the heating coils with a loss of only 10 to 15 per cent of its heat value in passing through the engine.

The writer is holding no brief for any type of kiln because he does not pretend to be intimately familiar with the drying of lumber. He would, however, be very much interested in an exhaustive and authoritative discussion of the merits of the various types of kilns as related to the drying of various kinds and thicknesses of lumber.

Kenneth Redman. In order that this discussion may bring to light the view points of those who are interested in other types of kilns exception must be taken to some of the broad general statements with which Mr. Perry dismisses the discussion of them.

The writer refers particularly to the blower or forced circulation type of kiln which it is inferred is suitable only for the drying of extremely thin stock or veneer.

As a matter of fact the fan type of kiln is today meeting with its most marked success in the drying of extremely heavy stock which heretofore it has not been possible to dry in the pipe coil kiln without the liability of enormous losses in the drying process. An examination of some of the fundamental facts to be considered in drying lumber will clearly show why this is possible.

In the first place, the writer heartily agrees with Mr. Perry that the ideal way to dry lumber is to keep the surface pores open and to draw the moisture out of the wood only at the rate that the moisture can transfuse from the center to the surface. In order to secure the maximum rate of such transfusion the outer pores of the wood must be kept open, and our experience confirms the theory that to pass comparatively large volumes of air so conditioned that each cubic foot will absorb only one or two grains of moisture is by far the most speedy and economical. In other words, the fan type of kiln does not attempt to saturate the air completely, thus causing it to cool, become heavy and drop and thus create a circulation.

This feature of the pipe coil kiln regardless of its particular type depends upon the varying density of the air in order to stimulate circulation, and consequently the drying power or moisture deficit varies directly as to the length of time that the air remains in contact with the lumber. This brings us to the point that the air should be kept in contact with the lumber for as short a period as is commercially practicable so that each individua board in the pile of lumber may be subjected to air of practically thle same drying power. The shorter the air travel the more uniformly dry will be the lumber. A striking example of this feature is the fact that Mr. Perry estimates that every cubic foot of air discharged from the pipe coil kiln picks up twenty-five grains of moisture. Granting that this figure is correct it is obvious that the lumber which is subjected to the air after it has absorbed nearly this amount is going to dry much more slowly than the lumber which is in contact with the air immediately after it leaves the heating coils. Contrast this procedure with that of the fan kiln where the air is properly conditioned before it enters the kiln and is removed from the kiln after it has picked up at the outside not over one or two grains per cubic foot. The uniformity of drying conditions under the latter system is obvious.

The principal duty of air in any drying process is to act as a

conductor of heat which is transferred to the material to be dried, thus causing the moisture to vaporize and then to be carried away. It is much more efficient to move air by means of a fan than to depend upon the heating and cooling effect of air. The writer would like to draw a simile of the importance of adequate circulation in drying lumber to that of a railroad system of a country delivering heat units to the lumber and carrying away in its place water vapor. If this so-called transportation system falls down or is inadequate the results correspond likewise.

In conclusion, the writer feels that it must be obvious that it is much more economical to recirculate and recondition the air to the desired point and to use it over and over again with very little expenditure of power than it is to take normal outdoor air, heat it to the desired point and then expel it, thus using the heat units only once. Furthermore, the fact that practically all thin veneer stock is dried almost in direct contact with hot steam pipes with a minimum of circulation, and the fact that some of the largest concerns in the country drying extremely heavy stock are using the fan type of kiln, would tend to offset Mr. Perry's statements that the fan type of kiln is applicable only to thin stock and that it is not advisable for drying 4- and even 6-in. lumber. While this might have been true twenty years ago before the value of drying with high humidities was known, it certainly is not true today when there are numerous installations using extremely high humidities and rapid circulation turning out the most difficult class of wood products with less than 1 per cent loss run after run.

The Author. Mr. Cutler's comments relate almost wholly to the theoretical advantages of rapid and positive circulation, rather than to proving that proper "strainless" drying can be safely and economically conducted in a rapidly moving atmosphere. The author does not controvert the statements that a blower kiln will dry surfaces rapidly, but does state, and has repeatedly proved in practice, that any rapid surface drying almost inevitably leads to serious damage to lumber. To prevent rapid surface drying the humidities must be high and high humidities are not attainable in blower kilns. To quote from the author's article, — "High humidity and large circulation of air are antithetical to one another. To obtain high humidity, the circulation must be either stopped altogether or greatly reduced, and to reduce the humidity a greater circulation must be induced by increasing the draft."

At one point Mr. Cutler speaks of a "speed of fourteen changes of air per minute." Taking the typical kiln in the author's article, in which the net cubic contents was 6,277 cu. ft., and multiply by 14, to ascertain the amount of air to be moved each minute, and we obtain 87,878 cu. ft. If the fan discharge is 10 sq. ft. area, the required velocity will be nearly two miles per minute! To reduce the velocity to a mile per minute would require a discharge area of 16 sq. ft.! This does not seem reasonable, and the power required for such an air movement would be exceedingly costly.

Perhaps the best argument for the writer's conclusion is that blower kilns, once much used in piano and furniture factories, have largely given place to ventilated kilns. With the rapid trend of better manufacturing methods in woodworking, this should carry considerable weight in the minds of those who are selecting types of lumber dryers.

No. 1879

EFFECTS OF LARGE SUDDEN STEAM DISCHARGES FROM BOILERS

REPORT OF RESEARCH SUB-COMMITTEE ON SUDDEN INITIAL POP LIFT OF SAFETY VALVES

THE Committee submitting this report was appointed about a year ago to investigate primarily a single, definite problem in connection with the use of safety valves as follows:

2 When a relatively large steam discharge is suddenly[1] released from a boiler, or such a discharge from a boiler is suddenly stopped as in the opening and closing of a safety valve, is there a shock to the boiler which, under any conditions, may become dangerous or cause damage to the boiler? If such a shock of sufficient magnitude is detected, would it seem wise to recommend limiting the capacity or lift of valves qualifying under the A.S.M.E. Boiler Code rules, and thus require more valves of smaller capacity per boiler rather than fewer valves of larger capacity than would be required if no limit were thus imposed?

3 Any shock, except that due to the actual impact of the small valve parts, such as the disk upon its seat must be transmitted to the boiler shell through the medium of either the steam or water contained in the boiler. A pressure gage, connected at different points in the boiler while the valve is blown, would thus be all that is necessary to measure the magnitude of the shock,

[1] Spring safety valves of present standard designs all have an equal suddenness of opening, although of course to different heights of lift. That is, on lift diagrams in which the abscissas represent time to the scale of two or three feet per minute, no difference in the verticalness of the opening line is measurable.

It is of course possible to modify this action with some dash-pot dampening device or its equivalent, and new designs may incorporate such auxiliary apparatus, but such designs are not general valve practice at present.

Contributed by the Research Committee for presentation at the Annual Meeting, New York, December 4 and 7, 1922, of The American Society of Mechanical Engineers. Received by the Council, January, 1923.

if the gage were sufficiently quick and accurate, i.e., operated virtually without inertia of parts.[1]

4 This report merely describes such a gage and the results obtained with it applied to different points in a boiler upon suddenly releasing and stopping varying discharges of steam under different known conditions.

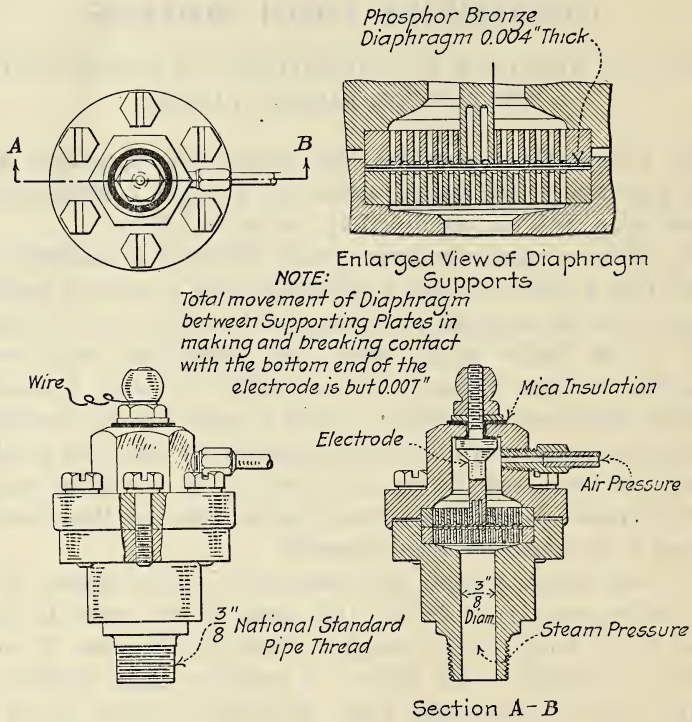

Fig. 1 Assembly of Pressure Indicator Used for Detecting Fluctuations of Boiler Pressure

5 Prof. Edward F. Miller, a member of this Committee, was able early in the year to interest three of his graduating students at the Massachusetts Institute of Technology into undertaking a

[1] The pressures in a gun barrel resulting from the explosion of the powder charge at its breach are thus measured and can certainly be considered to be as sudden and difficult of measurement as any action which takes place within a boiler. Gun-barrel pressures are determined by the compression of small copper disks by a plunger communicating with the interior of the barrel.

partial investigation of this problem as the subject of their thesis. These men were C. W. Greening, W. W. Harris, and J. K. Macomber; and a copy of their thesis is submitted, attached to this report.

6 The pressure-indicating device which was used in this thesis work had been developed in 1919 at the Bureau of Standards at Washington, D. C., for determining cylinder pressures in internal-combustion engines, and the whole arrangement of apparatus developed in conducting this thesis was practically duplicated in the more extended investigation recently carried on at the plant of the Consolidated Safety Valve Company at Bridgeport, Conn.

DESIGN OF PRESSURE INDICATOR

7 This device as used is shown in Fig. 1. It consists of a diaphragm mounted between two supporting plates, which plates are grooved and drilled as shown. The top supporting plate on its bottom surface, next to the diaphragm, is dished very slightly upward to give the diaphragm at its center a movement of 0.007 in. between the top and bottom supporting plates. The central portion of the top plate consists of a separate part, or electrode, which is mica-insulated from the rest of the indicator. When the diaphragm rests against the top supporting plate it touches this electrode, closing the circuit through it. When, on the other hand, the diaphragm is forced down against the bottom plate, the circuit is broken. The shallow concentric grooves and the holes through the plates to these grooves keep relatively full-area pressures against both sides of the diaphragm from the communicating passages at all times.

8 The bottom of the diaphragm communicates through the hole in the threaded shank of the indicator with the steam or water pressure in the boiler at the point of attachment. The space above the diaphragm is connected to a compressed-air line which is used to balance the diaphragm as described later. The only way in which this indicator as used at Bridgeport differed from that used in the thesis work above referred to, was in the substituting of phosphor-bronze for steel diaphragms, and in the use of mica for the insulation throughout; the former giving better electrical contact, and the latter being required by the greater heat at the boiler-tube applications.

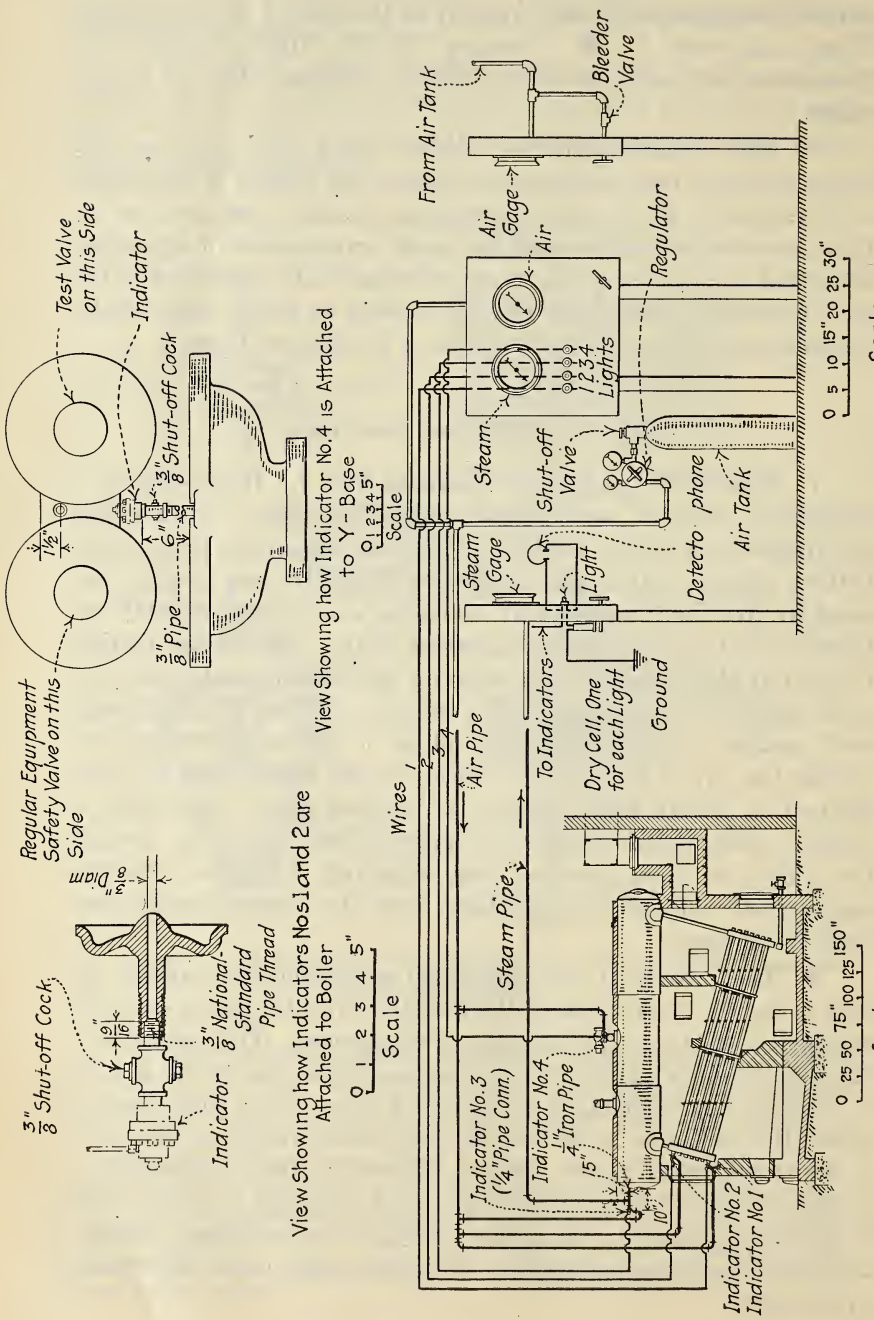

Fig. 2 Apparatus for Detecting Fluctuations of Pressure at Different Parts of a Boiler Under Different Conditions of Steam Discharge

TEST APPARATUS AND ITS ARRANGEMENT

9 Fig. 2 shows the apparatus as arranged in the Bridgeport tests.

10 *Pressure Indicators.* At several discussions among members of the Committee the points in a water-tube boiler at which a shock, if it occurs upon the sudden relief of steam, was considered to be most likely were indicated as follows:

a At the upper or front ends of the tubes where the water in circulating takes nearly a right-angled turn into the front header

b At the top of the front header where it enters the drum

c Against the headers of the drum

d At the middle of the drum.

11 Accordingly, as shown in Fig. 2, the indicators were placed as follows:

1 Opposite the upper end of the center tube of the lowest row, a $\frac{3}{8}$-in. hole being drilled through the bolt of the handhole plate, and a $\frac{3}{8}$-in. gate valve being inserted between the indicator and the bolt as shown in the separate detail

2 Similarly located opposite the upper end of the center tube of the top row

3 On the end of 10 in. of $\frac{1}{4}$-in. pipe leading horizontally into the front header 15 in. below its top edge

4 Vertically into the Y-fitting at the top center of the drum as shown in separate detail in Fig. 2. Upon the branches of this Y-fitting are the regular safety valve of the boiler, not blown during the tests, and the valve which was opened in testing, shown in Fig. 3.

12 It was not considered worth while to attach an indicator opposite the top end of the front header where it enters the drum, as this would have necessitated drilling or removing the baffle plate in the drum at this place and inserting an extension pipe through a hole and packing gland especially provided at the top of the drum.

13 *Steam Connections.* The attachment of these indicators as just described provided for a direct communication of the boiler steam pressure (in applications Nos. 3 and 4) and boiler water

pressure (in applications Nos. 1 and 2) to the under side of the indicator diaphragms. A line was run from the No. 3 indicator connection to a steam gage placed upon the test instrument board as shown.

14 *Compressed-Air Lines.* The part of each indicator above the diaphragm was piped into a single air line leading from the boiler room through a wall into the adjacent engine room where the hand operation was carried on and the observations made. This air line was attached to a large-sized oxygen tank especially filled with compressed air for these tests through the courtesy of Mr. G. O. Carter of the Linde Air Products Co., of Newark, N. J.

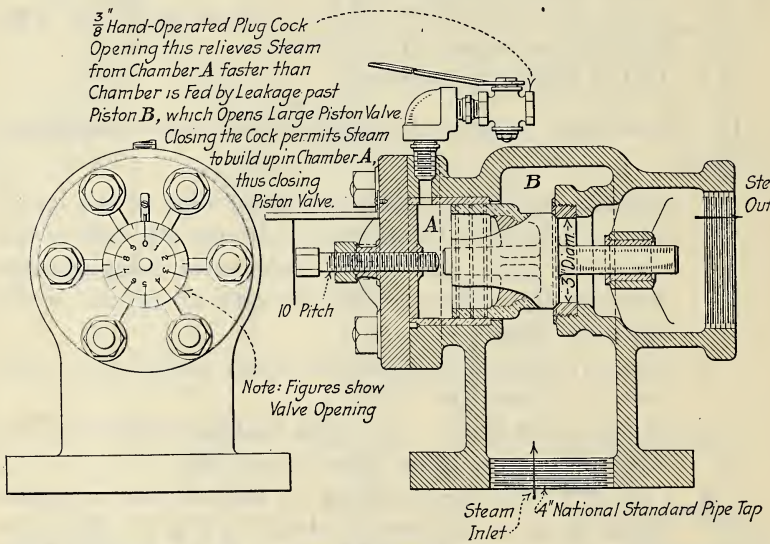

FIG. 3 ASSEMBLY OF HAND-OPERATED PISTON VALVE WITH ADJUSTABLE OPENING USED IN BOILER-PRESSURE TESTS

This air flask was equipped with a special high-pressure regulator for controlling the pressure delivered to the line leading to the indicators. The line was also connected to a large test gage calibrated in pounds, and to a needle valve used as a bleeder, as shown in Fig. 2.

15 *Electric Lines.* The bottom of each of the indicators was sufficiently grounded by its attachment to the boiler. The top end of each electrode was individually wired to a small $1\frac{1}{2}$-volt lamp mounted on the gage board. The other lamp connection passed through a single dry cell to a ground. With this arrange-

ment, when the indicator diaphragm was up against the top supporting plate, of which the lower end of the electrode formed the central part, the circuit was closed and the lamp lighted. When the diaphragm was against the lower plate, however, the circuit was broken and the light extinguished. In case a pressure should be applied and removed from the lower side of the diaphragm so suddenly that the filament of the lamp would not light, a detectophone was arranged to be plugged across any one of the four lines. With this the slightest contact of the diaphragm with the electrode could be clearly heard.

16 *Steam-Discharge Valve.* In the thesis tests, which were conducted upon a 75-b.hp. Almy water-tube boiler at the plant of the Crosby Steam Gauge and Valve Company at Boston, a number of safety valves of different sizes, pressures, and capacities were used to obtain the sudden release of steam. At Bridgeport, however, a special piston-operated valve, shown in Fig. 3, was provided. The amount of "lift" or horizontal movement of the disk of this valve in opening could be controlled by setting the 10-pitch stop screw, on which a graduated dial was provided to give a micrometer adjustment to 0.001 in. In this way the exact amount of the sudden steam discharge could be regulated from 0 up to a maximum of 98,800 lb. of steam an hour at 300 lb. gage pressure.

17 The action of this valve in opening and closing can be readily understood from Fig. 3 if the following two facts are noted:

a That the piston diameter of valve *B* is somewhat greater than the seat diameter, leaving a small unbalanced shoulder in the steam-inlet space of the valve

b That there is a slight clearance between the sides of piston *B* and the walls of chamber *A*, permitting full steam pressure to build up in chamber *A* from the leakage through this clearance. This leakage, however, is much less than the opening through the $\frac{3}{8}$-in. plug cock, which in conducting the tests was opened and closed by hand, so that when that is opened to atmosphere the pressure in *A* is relieved much faster than it can build up, and the valve *B* is opened by the steam pressure upon the small unbalanced shoulder referred to in Par. 17*a*.

18 The movement of this piston valve in opening and closing is so fast that when attached to the lift-recording instrument used

in taking lift cards of regular spring safety valves, it invariably bent the pencil arm, preventing the taking of a card.

19 *Test Boiler.* The boiler to which the indicators were attached and upon which the tests were run is a Babcock and Wilcox of the inclined-header, longitudinal-drum type, carrying 300 lb. pressure, rated at 94 hp., having 950 sq. ft. of heating surface and 30.9 sq. ft. of grate area. It has forty-nine 4-in. tubes 16 ft. 7 in. long and a 36-in.-diameter drum 21 ft. $2\frac{11}{16}$ in. long, the tubes being arranged seven rows wide and seven rows high.

20 The total safety-valve capacity for this boiler according to A.S.M.E. Boiler Code rules, requiring 6 lb. of steam per hour valve capacity per square foot of heating surface, is therefore 5700 lb. of steam per hour, which figure can be borne in mind in comparison with the actual steam discharges used during the tests.

METHOD OF CONDUCTING TESTS

21 The procedure was very simple. With all connections made as described above and the gate valves between the pressure indicators and the boiler open, the lamps on the observation board would all be lighted. This was because the steam pressure on the under side of all the diaphragms kept them pressed up in contact with the electrodes. The steam and air gages would then be checked together throughout their ranges by connecting both to the same air line, there being valves provided for this purpose although they are not shown in Fig. 2. The gages were then reconnected to their respective lines, the steam gage showing the pressure of steam in the boiler and the air gage, zero. By opening the air-tank regulator valve, air was admitted to the line leading to the top part of each indicator and to the gage. The lamps would remain lighted until the air pressure nearly equaled the steam pressure, and then, on increasing the air pressure, the lights would all go out, showing that the diaphragms had been forced down away from the electrodes in each indicator.

22 By raising and lowering the air pressure about this point it was noted that the pressure at which the lights would go out did not vary from that at which they would go on by over $\frac{1}{2}$ lb. pressure, showing a very sensitive response of the thin diaphragms to the pressure changes. However, this point at which the indicators would make and break contact did not always occur when the balancing air and steam pressures were exactly equal, the slight

excess of one over the other representing a very slight resistance of the diaphragm to its small 0.007-in. movement. This in no way reduced the accuracy of the observations since the actual contacting point was determined at the time of each test.

23 The running of a test merely consisted in carefully bringing up the air pressure until the lights went out, noting the pressures for each light, and then opening and closing the special discharge valve on the boiler. Any shock or increase in pressure, at any of the four indicator locations, due to this sudden opening or closing of the valve, would then have been shown by a relighting of that lamp or a click of the diaphragm as heard in the detectophone. The "lift" of the discharge valve was set for different relieving capacities and was blown at different pressures.

RESULTS

24 In none of the tests was any over pressure whatever noted, either by a relighting of one of the lamps or a click of the diaphragm as detected in the head 'phone. Hence the table of results (Table 1) need consist merely of an enumeration of the different conditions under which trials were made.

TABLE 1 RESULTS OF COMMITTEE'S TESTS OF SPECIAL PISTON-OPERATED SAFETY VALVE

Test No.	1	2	3	4	5	6	7
Lift of discharge valve, in.	0.2	0.3	0.4	0.5	0.6	0.7	full open
Steam-discharge rate, lb. steam per hour	30,500	36,400	47,100	59,600	73,200	87,500	95,000
Pressure blown at, lb. per sq. in.	282	282½	285½	286	287	288	288
Pressure closed at, lb. per sq. in.	281	281	284	284	285	285½	285
Air pressure to diaphragms, lb. per sq. in.	283	284	286	287	287½	288	288

NOTE. — Before and after these tests the air pressures required to make and break the diaphragm contacts, as shown by the lights, were tested and did not vary from the corresponding steam pressures by more than 2 lb. plus or minus as a maximum, the average variation being about 1 lb.

The hand-operated pilot valve was closed immediately after the main valve opened in each test, giving a low blowdown.

The water level was about half-glass and the boiler was steaming moderately in supplying steam to the valve-test room.

CONCLUSIONS

25 The results obtained both by the M.I.T. thesis men at the Crosby plant in Boston on a 75-hp. Almy boiler, and by our Committee at the Consolidated plant at Bridgeport on a 94-hp. Babcock and Wilcox boiler, were negative, i.e., they did not show under any of the conditions tried any degree of shock whatever.

26 These conditions covered sudden steam discharges which in size, relative to boiler capacities, were far in excess of the capacity of any safety valve which would ever be attached to the boiler. At Boston the maximum discharge tried was approximately three times the total safety-valve requirement of the A.S.M.E. Boiler Code, and at Bridgeport over seventeen times the total valve capacity for the boiler was instantaneously relieved in the attempt to produce a shock. When it is considered that two or more valves would be called for by the A.S.M.E. Boiler Code, the sudden steam relief tested at Bridgeport would be two or more times seventeen times the largest valve which would generally be used with this boiler.

27 The locations of the indicators in the Bridgeport test, selected because considered the most likely points to receive shock, are believed to be representative enough to justify the conclusion that no shock would be detected from an attachment at any other place.

28 This investigation might be continued with prolonged trials on different types of boilers, under different conditions of steaming, and with additional points of attachment for the diaphragm indicators. However, considering the fact that no degree of pressure increase or shock whatever has been detected under the severe steam-discharge conditions tried, we believe that it is extremely unlikely that any boiler conditions would be found so radically different as to produce a shock. In other words, if a pressure-increase shock could be produced by any reasonable boiler operation, we believe that the conditions of our tests were representative enough to have detected at least some trace of such shock.

P. G. Darling, *Chairman*
E. F. Miller
W. E. Jerauld
G. H. Clark
I. E. Moultrop
A. M. Houser
A. L. Fitch
G. S. Coffin

Committee on Sudden Initial Pop Lift of Safety Valves (Research Sub-Committee)

No. 1880

A.S.M.E. BOILER CODE

PART I—SECTION 5

Rules for the Construction of Miniature Boilers

M-1 *Definition.* Boilers to which the classification "Miniature" applies, embrace fired pressure vessels which do not exceed the following limits:

16 in. inside diameter of shell
42 in. length of shell
20 sq. ft. total heating surface
100 lb. per sq. in. maximum allowable working pressure.

Where any one of the above limits is exceeded, the rules for Power Boilers shall apply.

M-2 Specifications are given in these Rules, Pars. 23–178 of Part I, Section 1, of the Code, for the important materials used in the construction of boilers, and the materials for miniature boilers, for which specifications exist, shall conform thereto, except that in lieu of definite specifications for boiler plate material, there may be used for the shells or drums of miniature boilers, seamless drawn shells with integral heads, or seamless or extra-heavy lapwelded steel or iron pipe or tubing, provided it is of open-hearth material and the weld is formed by the forging process. Owing to the small size of the parts of miniature boilers, stamping as required by Par. 36 of the Rules for Power Boilers, need not be visible after completing the boiler, provided the manufacturer certifies on the data slip accompanying the boiler that the material is in accordance with the requirements of the A.S.M.E. Code for Miniature Boilers. Provisions shall be made by the manufacturer whereby he shall be able to supply complete information regarding the material and details of construction of any boiler built under the Miniature Boiler Code.

M-3 Steel plate, when used for any part of a miniature boiler where under pressure, shall be of the firebox or flange grades, but in

no case shall steel of less than $\frac{1}{4}$ in. thickness be used for riveted shells or less than $\frac{3}{16}$ in. thickness for seamless shells. The heads, if used as tube sheets with tubes rolled in, shall be at least $\frac{5}{16}$ in. thick.

M-4 The construction of miniature boilers, except where otherwise specified, shall conform to that required for power boilers. The factor of safety and method of computing the maximum allowable working pressure shall be the same as for power boilers.

M-5 Heads or parts of miniature boilers when not exposed to the direct action of the fire may be made of cast iron or malleable iron provided it complies with the requirements in Part 1, Section 1, of the Code for the headers of water-tube boilers.

M-6 Steam generator elements of not over 600 cu. in. in volume may be made of cast copper or bronze having a copper content of not less than 90 per cent and wall thickness of not less than $\frac{1}{4}$ in. Such generators shall be equipped with at least two brass wash-out plugs of not less than 1 in. iron pipe size and shall be tested to a hydrostatic pressure of 600 lb. per sq. in.

M-7 Circumferential reveted joints, where used, shall conform to the requirements in Par. 184 of Part I, Section 1, of the Code. Autogenous welding may be used for joints in miniature boilers where the strain is carried by other construction which conforms to the requirements of the Code and where the safety of the structure is not dependent upon the strength of the weld.

M-8 Tubes may be made of wrought iron, steel, drawn copper or drawn brass. Fire tubes $1\frac{1}{2}$ in. and over shall have both ends substantially expanded into the tube sheet by rolling and beading. Fire tubes less than $1\frac{1}{2}$ in. shall be expanded and beaded, or expanded and welded. The gage of the tubes shall not be less than that specified for water-tube boilers and fire-tube boilers as specified in Pars. 21 and 22 of Part I, Section 1, of the Code.

M-9 All rivet holes shall be drilled full size or they may be punched not to exceed $\frac{1}{8}$ in. less than full diameter and then drilled or reamed to full diameter.

M-10 The calking edges of plates, butt straps and heads shall be beveled to an angle not sharper than 70 deg. to the plane of the plate, and as near thereto as practicable. Every portion of the sheared surfaces of the calking edges shall be planed, milled or chipped to a depth of not less than $\frac{1}{8}$ in. Calking shall be so done that there is no danger of scoring or damaging the plate underneath the calking edge, or splitting the edge of the sheet.

M-11 Every miniature boiler shall be fitted with not less than three brass wash-out plugs of 1 in. iron pipe size, which shall be screwed into openings in the shell near the bottom, reinforced to give four full threads. All threaded openings in the boiler shell shall be provided with a riveted or welded reinforcement if necessary, to give four full threads therein.

M-12 Every miniature boiler shall be provided with at least one feed pump or other feeding device, except where it is connected to a water main carrying sufficient pressure to feed the boiler.

M-13 Each miniature boiler shall be fitted with feedwater and blow-off connections, which shall not be less than ½ in. iron pipe size. The feed pipe shall be provided with a check valve and a stop valve. The feedwater may be delivered to the boiler through the blow-off connection, if desired. The blow-off shall be fitted with a valve or cock in direct connection with the lowest water space practicable.

M-14 Each miniature boiler for operation with a definite water level shall be equipped with a glass water gage for determining the water level. The lowest permissible water level shall be at a point one-third of the height of the shell, except where the boiler is equipped with internal furnace, when it shall be not less than one-third of the length of the tubes above the top of the furnace.

M-15 Each miniature boiler shall be equipped with a steam gage having dial graduated to not less than one and one-half times the maximum allowable working pressure. The gage shall be connected to the steam space or to the steam connection to the water column, by a siphon tube or equivalent device that will keep the gage tube filled with water.

M-16 Each miniature boiler shall be equipped with a sealed spring-loaded pop safety valve, not less than ½ in. in diameter, connected direct to the boiler, independent of any other connection. The safety valve shall be plainly marked by the manufacturer with a name or an identifying trade-mark, the nominal diameter, the steam pressure at which it is set to blow, and A.S.M.E. Std. The minimum relieving capacity shall be determined on the basis of 3 lb. of steam per hour per square foot of boiler heating surface.

M-17 Each steam line from a miniature boiler shall be provided with a stop valve located as close to the boiler shell or drum as is practical.

M-18 Where miniature boilers are gas-fired, the burners used shall conform to the requirements of the American Gas Association,

as given in the Appendix. The burners shall in such cases be equipped with a fuel-regulating governor, which shall be automatic and regulated by the steam pressure. This governor shall be so constructed that in the event of its failure, there can be no possibility of steam from the boiler entering the gas chamber or supply pipe.

M-19 All boilers referred to in this section shall be plainly marked with the manufacturer's name, maximum allowable work-

A.S.M.E. Standard
Miniature
Serial No.....
.....lb.
.....
(Name of Manufacturer)

Sample of marking

ing pressure, which shall be indicated in arabic numerals, followed by the letters "lb.", and serial number. All boilers built according to these rules shall be marked A.S.M.E. Std. — Miniature. Individual shop inspection is not required for miniature boilers.

A data sheet shall be filled out for each boiler and signed by the manufacturer, this data sheet to include the most important items and to be numbered. In addition to this, the complete data sheet required for power boilers shall be filled out and preserved by the manufacturer for each lot of steel and each lot of boilers manufactured therefrom. The complete data sheet shall be marked to indicate to which boilers it applies and the manufacturer shall furnish copies of this complete data sheet when requested to do so by the owner of any one of the boilers. In requesting the complete data sheet the owner shall forward the number of the boiler which is stamped thereon in order that the manufacturer may readily identify the complete data sheet applying to the boiler.

Manufacturers' Data Report for Miniature Boilers As Required by the Provisions of the A.S.M.E. Code Rules

1. Manufactured by .....
(Name and address of the manufacturer)

2. Manufactured for .....
(Name and address of the purchaser)

3. Type ..... Boiler No. (.....) (.....) (.....) Yr. built
(Manufrs. Serial No.) (State and State No.) (A.S.M.E. No.) .....

Diameter of Length of

4. Shell or Drums.....Drums.....overall .....ft.....in.
(Inside of outside course)

Material for Shell, Straps,

5. Heads and Furnace Sheets made by .....
(If more than one make, give names of manufacturers in same order as parts referred to.

Has material used in boiler been checked for mill test reports..

6. Built for maximum allowable working pressure.....lb

7. Hydrostatic pressure applied.....lb.

8. Openings: No....Size...in., No....Size....in., No....Size...in.
(Main Steam connection) (Safety valve) (Blow-off)

Note: The mill test reports of tests of material used in this boiler are preserved by the manufacturer as well as all data applying to the boiler called for in the data sheet for Power Boilers. This data will be supplied by the manufacturer at the request of the owner of the Boiler.

.....
(Signature)

APPENDIX

Gas Burner Specifications — American Gas Association

Each burner shall be equipped as follows:

1 With a separate one-quarter turn gas cock.

2 With either an adjustable gas orifice or a removable brass orifice of a fixed drilling to meet the local condition.

3 With an adjustable air shutter capable of giving complete shut off; a lock washer or screw shall hold the shutter so securely that accidental shifting of the shutter is impossible.

4 The mixing tube shall be at least six times as long as its minimum diameter.

5 When the air mixer, mixing tube and burner are made in separate parts, they shall assemble so that there is no reduction in internal area at the point of their connection in the direction of the gas flow.

The burner proper should preferably be of a one-piece cored casting.

The port openings shall be drilled, or if assembled, shall be of uniform size.

The burner shall be capable of operating satisfactorily without a wire gaze.

For satisfactory operation a burner shall have sufficient flexibility to burn with a blue flame at full load and not flash back when shut down to the gas flow required to just maintain radiation losses.

A positive pilot-lighting burner shall be provided.

No. 1881

LIFE TESTING OF MECHANICAL INSTRUMENTS

WITH increasing large scale production of common measuring instruments, such as pressure gages, thermometers, ammeters, voltmeters and watt-hour meters, coupled with extensive use of such devices in connection with millions of automobiles, household utensils, and the like, in everyday use, — the matter of studying the life of such instruments and other more specialized sorts, becomes a matter of first importance. To double the life of an instrument made by the million annually, may be the means of saving capital charges and maintenance and field inspection costs enough to equip and man a considerable laboratory. Such testing frequently develops unexpected means of protecting an instrument from the effects of wear or abuse or of improving its service accuracy.

2 The purpose of life testing is obviously to predetermine, usually in the shortest practicable space of time, the character of service the device under investigation will afford after a stated period of actual use, and the probable extent of its useful life under the actual conditions of use, such conditions of use being approximated, so far as feasible, by setting up an average or normal manner of operation or "loading" or a representative extreme condition, depending upon whether the test is to determine the normal or minimum service life and performance. In the former case, the instrument may have to meet a requirement that under all specified conditions of service, an average service life of a certain period must be attained, while in the latter case, it may be required that under the most severe conditions of use or the most severe combination of such conditions, falling within the limits permitted by the specifications as to service, none of the instru-

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Report prepared by F. J. Schlink of the A.S.M.E. Special Research Committee on Permanency and Accuracy of Indication of Engineering Instruments.

ments shall fail. Failure may be defined to suit the requirements of the particular case in question, as complete effective cessation of the intended functions of the device, or quality of performance of the intended function falling without certain established limits of accuracy or sensitivity. For example, in the case of a stroke or revolution counter, failure may be determined as the point of life at which coördinated advance of the counting units definitely ceases (and such failure can normally be detected by simple observation) while in the case of a pressure gage, the indication of pressure, to be of value, must be at all times maintained within certain more or less restricted limits of accuracy. Naturally, the more complex the functions of the device, the more difficult it will be to set up testing conditions which rationally and correctly represent in a demonstrable manner, either average or extreme conditions of service.

SELECTION OF SPECIMENS FOR TEST

3 The proper selection of the instrument for test is most important; samples submitted by a manufacturer especially for examination are likely to be far from representative of the regular product, even though there is no definite intention to make a special selection of such samples. In general, it may be said that the cost of life testing of apparatus is high enough to warrant the purchase of test specimens on the open market, or their selection by a purchaser by random choice from the makers' stock shelves.

4 Nevertheless, even if samples known with certainty to be representative are not available, life testing may be proceeded with with reasonable assurance of developing valuable information, for the life test is pretty sure to bring to light hidden weaknesses and defects and to develop, by comparison with other instruments, points in which a given instrument is definitely superior.

PREPARATION FOR TEST

5 The test should be started with the instrument clean, and lubricated in about the manner that may be expected to characterize the regular stock of instruments of the same type at the time of their being put into service. It is practically impossible except in the case of continuously lubricated apparatus, and such are rare in the field of measuring instruments, to provide during a life test, lubrication of the same characteristics as that typifying service life. In the case of such an instrument as a time switch,

or a telephone calling dial, re-oiling in service may be specified, to be done at definite time intervals, say once every two years. In such cases, of course, re-oiling during life test would be done at intervals of operation equivalent to the number of operations expected to occur over the interval between oilings in normal use. For various reasons, this is only a coarse approximation to the desired result, since lubrication is affected by time, as well as by use, owing to drainage, oxidation, and probably also to an appreciable extent, to evaporation. Moreover, in rapid operation, the oil may be momentarily exhausted at a given bearing point and wear then proceed at a rapid rate, whereas in actual service, the long time intervening between some successive operations, may permit replenishment of the oil on the bare surfaces, by slow flow from neighboring oily areas.

6 There are a number of other respects in which an accelerated life test fails by a significant factor to approximate normal service, among which may be mentioned the effects of corrosion due to atmospheric exposure, complicated in many cases, as in the chemical industries, by the presence of corrosive gases and vapors in the air, or by high humidity; deposition of dust or gritty particles from the air, or from neighboring tools or machinery; fading or obliteration of graduations due to light or moisture; and what is perhaps most important practically, the abrasive and hence wearing effects of particles removed from rubbing surfaces, during operation. The deposition and distribution of these particles will be much different during accelerated test than in normal use, and it is a fact established beyond question that such abrasive particles affect the wear life of an instrument or small machine by a large factor. A case is in mind where the life of a simple contact-making device comprising a metallic brush rubbing on alternate segments of conducting and insulating material, was multiplied by perhaps five, by the simple expedient of dusting off the worn particles of metal from the brush and disc as fast as they appeared. In this case the abraded particles determined the wear; in others it may be the air-borne particles shed by near-by machinery or tools.

7 In general this points a very useful lesson, well supported in the writer's experience: that in any instrument in which the prevention of wear, and lengthening of service life are important, regular cleaning is essential; and more important still is the protection of the working parts, from air-borne dust particles, by use

of an airtight case, or one so designed, by the equivalent of baffling or otherwise, as to slow down the air stream entering the case so much as to occasion the dropping of the dust particles before the slow-moving air stream has reached the working parts.

8 Apart from the wearing out of working parts of the instrument occasioned by the entrance of dust, the errors of indication produced by increases of and changes in the friction are likely to increase to an important degree the need for and cost of recalibration. Instruments having a hinged door or cover should be so designed that the dust dislodged when the door is opened, does not fall inside the case. Apart from the cost of cleaning an instrument, better results are obtained if the dust is prevented from ever entering the working parts, as is well recognized in the care used in constructing watch and chronometer cases as nearly dust tight as possible.

9 The effect of dust on the wear of an instrument is so significant that it is to be recommended that in important cases, an attempt be made to approximate during the accelerated test the average dust deposition that may be expected to occur during normal use of the instrument.

DESIGN OF THE TESTING DEVICE

10 The test should be so carried out that abnormal deterioration of the instrument due to excessively high stresses, as those of impact, developed as a result of fast operation, can be avoided as far as possible. In this matter, as in those discussed above, it is to be remembered that an accelerated test can never correctly simulate service, but that a great deal can be done to eliminate obvious errors. For example, in life-testing of a pressure gage, the pressure can be applied quickly enough without introducing the impact effect of water hammer, even if a liquid medium is used for applying the pressure. Similarly in cam-operated mechanical testing devices, the design of a cam can easily be made to approximate the continuously varying acceleration of harmonic motion, in the absence of specific information regarding the nature of the acceleration to which the apparatus is subjected in use.

11 In some cases, as in automobile, locomotive, and airplane instruments, the definite introduction of vibration treatment of the instrument during test will be justified. As to the simulation of such vibratory service by artificial means, few data are available

to guide. A study of this subject in relation to airplane instruments, and of the effect of the shocks of transportation on the accuracy of aneroid barometers, was made at the Bureau of Standards before and during the war, under the direction of Professor M. D. Hersey; it is doubtful if there is any other experience in this field available for reference, although it is quite certain from experience with automobile clocks and locomotive pressure gages, that the impact stresses of such service greatly accelerate the wearing out of the instrument.

12 The establishment of the highest possible speed of operation of the instrument within the limit of reasonable simulation of service use is often a matter of importance, as in the case of some devices whose useful life is required to run into millions of operations; and in such circumstances the expenditure of considerable care and study in the refinement of the testing apparatus will well repay the effort in the time and supervision saved on the test, and the more prompt availability of results.

13 Another point regarding the details of the design of the testing machine relates to the difficulty already discussed, of rapid wearing out of instruments due to abraded metallic particles. There is great danger that unless special precautions are taken against it, the testing machine itself will contribute largely to the supply of such particles, and care is to be taken to protect the instruments under test by suitable screens and guards at all points at which they are subjected to bombardment of material including lubricating oil, discharged from the parts of the testing machine.

14 In the case of instruments which have a very long life, running into millions of operations, it will be impossible probably to design the testing machine so that it will outlive in every detail the instruments tested, and in such cases care should be given to devising the details of the machine for easy replacement at a minimum cost, often at the expense of compactness, and neatness of appearance.

MAINTENANCE DURING TEST

15 It is often impossible on account of the excessive cost involved, to test enough instruments of a given type to afford a true statistical picture of the performance of the type; and in but few cases would so exhaustive an examination be warranted. Fortunately, the differences between commercial instruments are of

a large enough order that normally no means of fine discrimination are required.

16 Thus, it is often desirable, as soon as a part breaks or a serious defect or error develops, to repair the fault and restore the instrument to test, due care being taken to record the stage of operating life at which each such failure takes place. A careful examination of the whole instrument should be made and noted at such times in order that the relation of the defect to others already or subsequently developing, may be determined in reviewing the data at the completion of the test, as it often happens that the occurrence of one type of failure will influence the appearance of another.

17 As has already been indicated, cleaning and re-oiling at proper intervals during the life test will need to be carried out, in most cases, and the record should include a statement as to the means of cleaning, whether wiping, brushing, blowing out, or washing with a solvent; and of the kind and amount of lubricant used.

CALIBRATION DURING AND AFTER TEST

18 The ordinary type of calibration will not be of great service in this sort of investigation, except with instruments of relatively low variance. Hysteresis or cyclic calibration conducted under the régime laid down in the several papers on this topic will aid definitely, however, in showing up important differences in amount of wear, failure of lubrication, etc. For a fuller discussion of this point see the three articles listed below,[1] wherein the subject of friction and backlash in their relation to instrument errors are discussed in detail, and it is shown that the hysteresis or load-deflection loop of an indicating instrument operated aperiodically over a definite range, after cyclicization, and in the absence of external vibration, affords a powerful means of discriminating between types and individuals. In the wear test, these methods are of especial value, because they disclose the net effective result of the wearing action, integrating as it were the wear damage of all the journals, bearings, pivots, gears, rollers, cams, etc., involved in

[1] The Concept of Resilience with Respect to Indicating Instruments, *Journal Franklin Inst.*, Feb., 1919, vol. 187. p. 147. The Determinateness of the Hysteresis of Indicating Instruments, *Journal Wash. Acad. of Sciences*, Jan. 19, 1919, vol. 9, p. 38. The Calibration Function in Indicating Instruments, *Engineering* (London), Aug. 15, 1919, vol. 108, p. 204.

the linkwork, as well as all resulting changes in frictional resistances, whether due to change in the finish of surfaces, exhaustion of lubricant, or interposition of dust particles. Means are readily developed for applying the same or equivalent technique to integrating or value-controlling instruments. The hysteresis determination will afford a better index of the state of the instrument before, during and after a given wear life than the other available means of estimation such as calipering and weighing of journal and bearing parts; this because the cyclic calibration method automatically and necessarily weights each component of wear in proportion to its proper part in determining the inaccuracy and variance of the indications of the instrument.

19 Of course, the cyclic calibration need not be carried through in the presence of such massive and perhaps fatal defects, as breakage of parts, marked sticking or jamming of joints, or significant derangement of important adjustments.

20 Other means of interpreting wear damage are familiar, as calipering of holes and journals, and weighing of parts that have relative motion, to determine the amount of material removed. The clearance of a journal in its bearing can be conveniently determined by observation with a low-power measuring microscope, or more simply by moving one part back and forth in relation to the other while the plunger tip of a dial micrometer is kept in contact with the moving element. The amount of clearance can be readily estimated from the maximum amplitude of the indicating hand of the gage. The depth of shallow pits, or depressed thrust areas can be measured by a depth-measuring microscope.

THE NATURE OF MECHANICAL WEAR

21 A very important and suggestive study of the wear of metals which run together is found in a paper by L. Jannin and L. Guillet. (*Rev. de Met.* 19, pp. 109–116, and pp. 117–119, Feb. 1922.) These investigators found, under the conditions of their experiment, which permitted ready measurement of the worn surface, that the principal cause of wear in bearings is insufficient polish on the rubbing surface of the journal, while the chief cause of the wear on the journal is impurity in the lubricating oil. The latter condition, no doubt, will result from impurities introduced in the form of dust or gases by exposure to the atmosphere or by imperfect cleaning of the bearing parts at the time of manufacture. The best results as to endurance of the bearing were obtained not

when the surfaces were of materials of maximum dissimilarity, but when both consisted of a cemented and quenched case-hardening steel.

22 It is probable that instead of a minimum of wear being determined by dissimilarity merely, of surfaces, fineness of grain in both surfaces might produce the same result. Perhaps the maximum wear occurs when the maximum projection of particles transverse to the direction of rubbing exists, so that when a particle is torn out by abrasion, its uprooting leaves the surface again relatively rough.

23 When a small particle is uprooted by the abrasive action in the bearing, it leaves the smoothness and mode of frictional engagement of the surrounding surfaces relatively unchanged; when, with coarse structure a large particle is uprooted, its removal results in increased wear for a time, due to the shift of the directions of the particular elementary reactions between the wearing surfaces, and the exposure of new, hitherto protected, particles to the wearing action.

LIMITATIONS AND SPECIAL FEATURES OF THE FORCED TEST

24 It seems likely that even under the best conditions, the forced test cannot be applied without making due allowance, in the interpretation of results, for the specific points in which such tests cannot quite represent service conditions. In some cases the forced test is less severe than service, especially, for example, in cases where the movements in the instrument are mainly elastic rather than mechanical; but usually the forced test is rather more severe than the normal service. Nevertheless, due to the fact that some commercial instruments fall far short of normal requirements or differ so widely in a given type that discrimination is easy, the forced test is destined to play a more and more important rôle in instrument investigation and control. Certain it is that if makers of cheap watches were to have subjected samples to life tests under moderate jolting, at a balance vibration period artificially adjusted to say one-fifth the normal running value, many serious defects which characterize these watches would have been corrected before they were so widely marketed. Similarly, certain indicating and recording thermometers and pressure gages are subject to a very serious and troublesome shift of the zero due to uncoiling of the Bourdon tube easily brought to light by life test conditions.

25 It is to be noted that in a good many cases, the life test

of an instrument may be made self-recording, or self-limiting, so that failure to register correctly may either be made to stop the test, or to register itself in some easily interpreted way. For example, as described in outline in the paper by the writer in *Management Engineering,* June, 1922, a considerable number of recording thermometers, or pressure gages may be simultaneously life-tested in such a manner that the occurrence of important discrepancies can be directly noted on the record chart, by comparison with a reiterated standard temperature or pressure. Similarly, recording voltmeters and ammeters may be made to write their own life history by actuation through proper regulating resistances from an interrupted source of constant potential. Many electrical instruments, speed-governing devices and the like may be subjected to constant automatic supervision by connection to voltage release or time-interval relays. Pressure regulators may be "supervised" by a recording pressure gage.

CONCLUSIONS

26 The forced test method of studying a product, whether looking toward improvement, re-design, or competitive quantity purchases, is an important technique, especially meriting intensive development in the field of measuring instruments, where new methods may be successfully generalized, in most cases, to a wide range of types of apparatus. This method of test can go far in eliminating the uncertain and always debatable experience factor in instrument design, and will ultimately place design for production upon the same stable basis as design for research. Development of improvements and new devices will not then have to await the slow trying out of performance in service, with the unreliable and conflicting types of information which field observations normally imply, and the fact that such information does not commonly reach the people whose decisions control design. It is easy to ascribe failures in the field to abnormal conditions or abuse; in laboratory tests the conditions, even though they can never precisely represent service, can be standardized and controlled with certainty and flexibility. In the matter of dust-tightness, for example, such differences as may appear in ordinary inspection of installed instruments are too easily ascribed to accidental differences in severity of exposure as affected by air currents, fineness of dust particles and other factors that are highly variable under service conditions.

No. 1882

NECROLOGY

ALBERT CARTER ASHTON

Albert Carter Ashton, general manager and treasurer of the Ashton Valve Co., Boston, Mass., died on January 31, 1922, in St. Petersburg, Fla. Mr. Ashton was born on May 12, 1869, in Norfolk, England. He attended the Chauncey Hall preparatory school in Boston and later the Massachusetts Institute of Technology from which he was graduated in 1889.

He served his apprenticeship with the Ashton Valve Co., a concern which was founded by his father. In 1892 he was appointed superintendent and two years later became secretary, assuming the additional duties of treasurer in 1895. For the last twenty years, Mr. Ashton has held the positions of treasurer and general manager of the company.

Mr. Ashton became a member of the Society in 1901. He was also a member of the New England Association of Commercial Engineers, the Employers' Association of Eastern Massachusetts, the National Metal Trades Association, the Boston Chamber of Commerce and the Somerville Board of Trade. He belonged to the Masonic Order, was a Knight Templar and a member of the Mystic Shrine. He belonged also to a number of clubs, including the Engineers' Club of Boston and the New England Railroad Club.

ELTON D. BAILEY

Elton D. Bailey, who became a member of the Society in 1912, died on March 29, 1922. Mr. Bailey was born on June 16, 1877, in Richmond, Mich., where he received his grammar and high-school education. Later he took the three-year mechanical engineering course at Michigan Agricultural College. Upon his graduation he became actively engaged in cement-plant construction and operation, having charge of the power equipment of Portland cement plants at Cement City, Mich., Durham, Ontario, Rockmart, Ga., Independence, Kan., Eagleford, Tex., and Macon City, Iowa.

During the War he was associated with the Packard Motor Car Co., Detroit, Mich., as engineer in the construction and maintenance department. Late in 1921 he returned to his home in Richmond, Mich., where he died.

JOHN A. BENNIE

John A. Bennie, assistant in the planning department of the Ohio Body & Blower Co., Cleveland, Ohio, was accidentally killed on October 18, 1922. Mr. Bennie was born on March 29, 1888, in Chatham, Ont., Canada, where

he received his early education. Later he attended Queens University, from which he was graduated in 1916 with the degree of M.A.

For about a year Mr. Bennie was connected with the Western Electric Co., in the cost-reduction and tool-capacity section, and at the end of that period he entered the requisition section of the engineering department of the Emergency Fleet Corporation. Later he was for short periods with the Industrial Works, Bay City, Mich., in charge of time-study, and with the Nordyke & Marmon Co., Indianapolis, Ind., in the same capacity. In January, 1920, he became production manager for the Ohio Body & Blower Co., and in April, 1922, assistant in the planning department.

Mr. Bennie became an associate member of the Society in 1922.

JOHN ANDERSON BENSEL

John Anderson Bensel, prominent consulting engineer of New York City, died on June 19, 1922. Mr. Bensel was born in New York City on August 16, 1863. He received his engineering training in Stevens Institute of Technology, where he was graduated in 1884 with the degree of M.E.

His first work was with the Pennsylvania Railroad as rodman. Later he worked on the Croton Aqueduct for a short period and then entered the employ of the New York City Dock Department, where for some years he was assistant engineer in charge of the North River waterfront. He resigned from this position in 1895 to open his own office, and the following year acted as engineer for the Girard estate and the city of Philadelphia in the improvement of the waterfront and the widening of Delaware Ave. along the Delaware River. He also acted as engineer for the city of Newburgh in fixing the tax valuation of the property of the Pennsylvania Coal Co.

In 1898 he became chief engineer of the Department of Docks and Ferries of New York City, which post he held until 1906 when he was appointed commissioner for two years. During this ten-year period the greater part of the water-front improvements of the city were designed and constructed, notably the large piers immediately below Twenty-second Street where the North River was deepened to accommodate the largest steamers.

From 1908 to 1911 Mr. Bensel was president of the Board of Water Supply in charge of building the Catskill Reservoir and Aqueduct. In 1910 he was elected State Engineer and held that position until 1914, during which time the greater part of the State Barge Canal was constructed. He was engaged in private practice when the War called him away and he entered the Army with the rank of major in the Engineers' Corps. He had charge of the construction work at the Naval Base in Norfolk, Va., in connection with the embarkation of troops for the War zone and also work in connection with Norfolk's water supply, which threatened at one time to become inadequate. Later he had command of the 125th Engineers at Camp Forrest, Ga. He was mustered out of the service in January, 1919, and returned to his private practice.

Since then he has been engaged as engineer in various special projects — engineer for the cities of Schenectady and Oswego in bridge and hydro-

electric development work; engineer for the Port Improvement of Jersey City, N. J., member of the Board of Consulting Engineers of the New York State Bridge and Tunnel Commission, etc. At the time of his death he was engaged in some large valuation work for the City of New York.

Mr. Bensel was a member and a past-president of the American Society of Civil Engineers, a member of the Institution of Civil Engineers of Great Britain, of the Institute of Mining Engineers and the Society of Military Engineers. He became a member of our Society in 1917. He was also a member of the Municipal Art Society of New York and of the U. S. Pan-American Society. He belonged to the Engineers', University, Union League and the Army and Navy Clubs as well as to several country clubs. In June, 1922, Stevens Institute conferred upon him the degree of Doctor of Engineering.

ERNEST WILKINSON BENTLEY

Ernest W. Bentley, consulting engineer, Jacksonville, Fla., died on August 14, 1922. Mr. Bentley was born on November 14, 1872, in New Brighton, Pa., where he received his early education. Later he entered Cornell University from which institution he received his M.E. degree in 1894.

Upon graduation he served his apprenticeship with the firm of Bentley & Gerwig, Ltd., then becoming connected with the Pierce-Crouch Engine Co. From 1896 to 1900 he held the position of superintendent with the Champion Saw & Gas Engine Co., Beaver Falls, Pa. He resigned from this concern to become superintendent of the Braddock Machine & Manufacturing Co., Braddock, Pa., where he was located for about a year when he became associated with the Carnegie Steel Co., in the engineering department. In 1903 he became sales manager of the Rust Boiler Co. and the following year manager of the Pittsburgh office of the Skinner Engine Co.

Mr. Bentley was compelled to resign from this position in 1916 because of ill health and he went to Florida where he bought and worked a farm. Regaining his health, he opened his own office in Jacksonville, Fla., as a consulting engineer on machinery. He was engaged in this work at the time of his death.

Mr. Bentley became a member of the Society in 1904. He was also a member of the Florida Engineering Society and a number of organizations and clubs in Rochester, Pa.

AXEL E. BERGGREN

Axel E. Berggren, assistant professor of steam and gas engineering in the University of Wisconsin, died on August 10, 1922, from injuries received in an automobile accident. Professor Berggren was born on October 16, 1882, in Marshalltown, Iowa, where he attended grammar and high school. He was graduated from Iowa State College in 1908 with the degree of B.S. in mechanical engineering. In 1914 he received the degree of M.E. from the University of Wisconsin.

Upon graduation Professor Berggren became an instructor at the University of Colorado where he remained until 1910, when he accepted an instructorship in the University of Wisconsin. In 1917 he was appointed assistant professor of steam and gas engineering in charge of the steam and gas laboratory of the University, which position he was holding at the time of his death.

Professor Berggren became an associate member of the Society in 1915 and was advanced to full membership in the spring of 1922. He was a member of the Society for the Promotion of Engineering Education and of several honorary as well as social fraternal bodies.

WILLIAM NEWTON BEST

William Newton Best, one of the very well-known members of the Society in the field of liquid-fuel burning, died in Brooklyn on April 11, 1922. Mr. Best was born at Clayton, near Quincy, Illinois, on June 3, 1860, and attended the public schools there and the Gem City Business College. He then went into railroad work and continued in various capacities until 1901, when he resigned his position as superintendent of the Los Angeles and Salt Lake Railroad to begin the manufacture of his own inventions. At the time of his death he had U. S. letters patent covering forty-four devices, most of them relating to the oil-burning industry, and was president and consulting engineer of the W. N. Best Furnace and Burner Corporation at 11 Broadway, New York.

Dr. Best was the author of the book Science of Burning Liquid Fuel, and in 1917 the honorary degree of Doctor of Science was conferred on him by the Lincoln Memorial University, Cumberland Gap, Tenn., of which he was for some time a member of the Board of Trustees.

He became a member of the A.S.M.E. in 1912; he was one of the committee appointed in 1918 to revise its Boiler Test Code, and served for some time as its representative on the Board of the Engineering Societies Library. He was also a Fellow of the Royal Society of Arts, London, and a member of the American Institute of Mining and Metallurgical Engineers, the American Institute of Metals, the International Railway Fuel Association, American Railway Master Mechanic's Association, The Franklin Institute, the Aero Society of America and of the New York Academy of Sciences.

He was also a member of the Long Island Lodge 382 F. and A. M., vice-president of the Board of Directors of the Goodwill Industries of Brooklyn, Inc., vice-president of the Board of the Williamsburgh Rescue Mission, Inc., and a member of the Board of Governors of the Neponsit Club of Long Island.

Dr. Best will be missed by a host of friends, as well as in the many activities in which he was interested.

CHESTER BORGSTROM

Chester Borgstrom, draftsman with the W. A. Jones Foundry & Machine Co., Chicago, Ill., was accidentally shot and killed on April 5, 1922. Mr. Borgstrom was born on September 8, 1899, in Chicago, Ill., and received his

education in the schools of that city, attending the Lewis Institute and later the Armour Institute of Technology. He was connected with the W. A. Jones Co. for five years, serving his apprenticeship and learning his trade with that firm.

Mr. Borgstrom became a junior member of the Society in 1921. He was a member of the American Legion.

WILLIAM H. BRADLEY

William H. Bradley, for 36 years chief engineer of the Consolidated Gas Co., New York City, died suddenly on January 18, 1922, of heart disease. Mr. Bradley was born in August, 1838, in New Haven, Conn. He was formerly chief engineer of the Municipal Gas Co., New York City.

Soon after he entered the employ of the Consolidated Gas Co., Mr. Bradley made an extensive tour of Europe, returning with a detailed report on all the foreign holdings of the company and plans for the future guidance of the concern in their development in New York City. He is credited with the discovery of the process of enriching coal gas with oil for illuminating purposes. His experiments in this direction were successful and later he developed what is known today as carburetted water gas. He designed the Astoria, L. I., plant of the company which he formally opened about ten days before his death. This plant is said to be the largest of its kind in the world, having a 15,000,000 cubic-foot gas holder.

Mr. Bradley became a member of the Society in 1892. He was a past president of the American Gas Institute and a member of several other technical gas associations. He belonged to the Masonic fraternity.

CARL F. BUCK

Carl F. Buck, who became a member of the Society in 1914, died on December 1, 1922, at his home in Newark, N. J., having been ill for over a year. Mr. Buck was born on June 8, 1871, in Coffeyville, Kansas. In his youth he learned the machinist's trade, later taking up architectural designing in which he was actively engaged for twelve years in the cities of St. Louis, Kansas City and Chicago.

His leaning toward mechanical engineering led to his forming a connection with the American Smelting and Refining Co., with which firm he continued with but few interruptions until his death. In his capacity as mechanical engineer and designer he was closely identified with the design and construction of the company's most important properties, particularly the smelteries at Murray and Garfield, Utah, and Hayden, Ariz. By virtue of his many years of service he was possessed of a knowledge of the several plants' requirements which was equalled by few of his associates.

Mr. Buck was an inventor of some ability and owned a number of patents for various mechanical devices. At the time of his death he had been a member of the Society for eight years. He belonged to the Masonic Order.

ELLIS M. BURR

Ellis M. Burr, president of the Burr Company, Champaign, Ill., died on April 3, 1922. Mr. Burr was born in Woodstock, Ill., on February 27, 1858. He was graduated from the University of Illinois in 1878 with the degree of B.S. and immediately opened a machine shop in partnership with Mr. Robinson in Champaign. The partnership was of but few years duration but the work was carried on by Mr. Burr individually until 1912 when the corporation of the Burr Company was formed.

Mr. Burr's accomplishments were largely in the precision field, where he was responsible for the invention and development of thermometer graduators, bottle graduators, bottle markers, tractor dynamometers, railroad dynamometers, etc. During the War the firm manufactured and furnished the special structural work for all but two of the Aviation Fields, ammunition hangfire testing equipment, a special dynamometer for testing Liberty motors in flight and also did a large amount of work for the Emergency Fleet Corporation.

Mr. Burr had practically retired from active business, having been in ill health for about two years. He became a member of the Society in 1917. He was a member of many local and civic clubs in Champaign and also of the Hamilton Club in Chicago.

WILLIAM CAIRNS

William Cairns, who, until his retirement in November, 1920, was vice-president and general manager of the Detroit Pressed Steel Co., Detroit, Mich., died in St. Petersburg, Fla., on December 19, 1922. Mr. Cairns was born in Charlottetown, Prince Edward Island, Canada, on August 17, 1865. He was educated in the Charlottetown schools and the Boston Technical School.

He served his apprenticeship as machinist for three years with Geo. G. McLaughlin, Boston, Mass. From 1884 to 1887 he worked in the tool room of the Walworth Manufacturing Co., So. Boston, then becoming tool maker with the Thompson-Houston Electric Co., Lynn, Mass. Five years later he became machine-shop foreman of the Boston Cash Register Co., Northampton, Mass. From 1892 to 1895 he served as tool maker with the Overman Wheel Co., Chicopee Falls, Mass. At the end of that time he became superintendent of the Spaulding Machine Screw Co., Buffalo, N. Y. From 1898 to 1900 he was superintendent of the Monarch Bicycle Co., Chicago, Ill., resigning to become master mechanic for the Howe & Path Co., Danville, Pa. In 1902 he became associated with the Elyria Iron & Steel Co.., Elyria, Ohio, as general superintendent, erecting a new mill for the company and designing and patenting a process for tube making. In 1905 he became development engineer for the Cadillac Motor Car Co., Detroit, Mich., resigning the following year to become superintendent of the General Railway Signal Co., Rochester, N. Y. In 1908 Mr. Cairns became general manager of the Parish & Bingham Co., Cleveland, Ohio. Three years later he reorganized the company, and under his management the concern expanded into one of the largest manufacturers of automobile frames in the country. He re-

signed from this firm in 1914 to become vice-president and general manager of the Detroit Pressed Steel Co., manufacturers of automobile frames and heavy steel stampings. He retired from active business on November 1, 1920.

For many years Mr. Cairns was a prominent figure in the automobile industry, his research work in pressed steel winning him wide recognition. He became a member of the Society in 1911. He belonged to a number of clubs in Detroit and Cleveland and was a member of the Elks and the Knights of Columbus.

CHARLES W. CASE

Charles W. Case, designer of special machinery for the Louisiana Celotex Co., Marrero, La., died on April 18, 1922. Mr. Case was born in New Orleans, La., on April 22, 1879. He received his early education in the public schools of that city and later his technical training in the Home Institute and the Tulane University of Louisiana Night School. He served for a short period as an instructor of mechanical drawing at the Institution.

He was well and favorably known throughout that section as a draftsman and mechanical engineer of ability, being at various times connected with the following engineering firms: Boland & Schwind Co., Ltd., chief draftsman; Ford, Bacon & Davis, chief draftsman; J. H. Menge & Son, machinery designer; John H. Murphy Iron Works, machinery designer; American Brewing Co., mechanical engineer; Wm. Garrigue & Co., chief draftsman; Freeport & Mexican Fuel Oil Corporation, superintendent of construction; Atlantic Loading Co., mechanical engineer; Ophuls, Hill & McCreary, superintendent of construction. His last position was with the Louisiana Celotex Co. as a designer of special machinery for paper manufacture.

Mr. Case became an associate-member of the Society in 1919.

FRANK WOOD CASLER

Frank Wood Casler, president of the Frank W. Casler Coal Co., New York City, died on October 27, 1922. Mr. Casler was born in Champaign, Ill., in June, 1867. He received his early education in the schools of that city and later attended the University of Michigan.

In 1892 Mr. Casler became chief engineer of the Newport Illuminating Co., Newport, R. I., in charge of operating. Eight years later he became associated with the Rhode Island Co., Providence, R. I., as chief engineer in charge of operating the Eddy and Manchester Streets power stations. After seven years with this company he became chief engineer of the Boston Elevated Railroad Co., in charge of operating the Lincoln Wharf and East Cambridge power stations. In 1910 he became connected with the Public Service Electric Co. of Newark, N. J., as general superintendent in charge of production. In 1919 Mr. Casler established his own coal business in New York City in which field of work he was engaged at the time of his death.

Mr. Casler became a member of the Society in 1913. He was also a member of the New Jersey State Board of Boiler Rules. He belonged to a number of fraternal organizations of Newark, including the Masonic Order.

ALFRED CLARKE

Alfred Clarke, well known as an inventor and the engineer who installed the Lick Observatory, died on April 27, 1922. Mr. Clarke was born in Leicester, England, on June 4, 1849. He was educated in the public schools of Leicester and in the department of science and art, South Kensington, London.

He came to the United States in 1876 as chief engineer of the Bradley Fertilizer Co., North Weymouth, Mass. From 1877 to 1885 he was connected with the Kitson Machine Co., Lowell, Mass., as superintendent and while there invented improvements in cotton machinery which are now used in every cotton mill in the world. For two years he was general manager of the Prospect Machine & Engine Co., Cleveland, Ohio.

Later in association with Arthur E. Childs he founded the Light, Heat & Power Co. of Boston which led to the formation of the Massachusetts Lighting Companies, owning twenty-seven gas, electric light and power companies in that state, of which Mr. Clarke was president when he retired seven years ago.

Mr. Clarke was a director of the American Investment and Securities Co., the Columbia National Insurance Co. and the Massachusetts Trust Co. He was one of the early members of the Society, joining the organization in 1886. He was also a member of the American Association of Engineers, the Boston Engineers' Club, the University Club of Boston and of many others, including the Masonic Order.

EDWARD E. CLAUSSEN

Edward E. Claussen, mechanical engineer, Hartford, Conn., died on August 15, 1922. Mr. Claussen was born on August 20, 1858, in Terre Haute, Ind. He was educated abroad, attending the Polytechnic Institute of Zurich, Switzerland. Upon his return to this country he engaged in the designing of labor-saving machinery in the paper, pulp and metal-goods industries. He was associated for seven years with the Continental Paper Bag Co. and from 1911 to 1921 was with the Public Service Cup Co., Brooklyn, N. Y., where he designed and built machinery for the manufacture of Lily paper cups. He also acted as consulting engineer for this firm.

Mr. Claussen had taken out over one hundred patents in the United States and abroad, covering his many inventions. He became a member of the Society in 1921. He belonged to the Masonic Order.

JAMES B. CLOUDSLEY

James B. Cloudsley, who at the time of his retirement in 1916 was chief engineer of the Bureau of Water for the city of Buffalo, N. Y., died on December 11, 1922. Mr. Cloudsley was born on March 26, 1848, in Dundee, Scotland, where he received his early education.

He served his apprenticeship with Charles Parke & Son, engineers of Dundee, and in 1869 came to the United States where for several years he worked as a machinist with the Nashville & Chattanooga Railroad. He

then went to Toronto, Ont., Canada, and became connected with the Grand Trunk Railway, later becoming inspector for the Canadian Steam Users Insurance Association. In 1887 he was made chief engineer for the National Transit Co. with offices at Williamsport, Pa. He resigned from this position in 1894 to accept the appointment of chief engineer of the Buffalo General Electric Co., Buffalo, N. Y., serving for six years or until the advent of the Niagara Falls electric power. In 1902 Mr. Cloudsley was offered and accepted the position of chief engineer of the Bureau of Water for the city of Buffalo from which he retired in 1916.

Mr. Cloudsley became a member of the Society in 1892.

FREDERICK W. COX

Frederick W. Cox, commercial engineer with Andersen, Meyer & Co., Ltd., Shanghai, China, died in 1922. Mr. Cox was born in November, 1864, in Summerton, England, where he received his early education.

He came to this country and served his apprenticeship with the Chicago & Northwestern Railway, Chicago, on locomotive construction, and was then assigned to the roundhouse of that line. A short while later he was appointed general foreman and master mechanic of the Chicago, Milwaukee & St. Paul Railroad, his work taking him to Babcock, Sioux City and Milwaukee as well as Chicago. His next position was as machine inspector with the Baltimore & Ohio Railroad at Baltimore, Md., from which he resigned to become superintendent of the Milwaukee Electric Co., Milwaukee, Wis. In 1910 Mr. Cox became assistant manager of works for the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa., where he remained for about three years when he became general salesman for Shewan Tomes & Co., Hongkong, China. About 1916 he became connected with the firm of Andersen, Meyer & Co., with which he was associated at the time of his death.

Mr. Cox became a member of the Society in 1908.

OLIVER CROSBY

Oliver Crosby, managing partner of the American Hoist & Derrick Co., St. Paul, Minn., and widely known inventor and manufacturer, died on December 8, 1922, of anemia. Mr. Crosby was born on January 29, 1856, in Dexter, Me., where he received his early education. Later he attended the University of Maine being graduated in 1876 as a mechanical engineer.

In 1882 he opened in St. Paul a small machine-shop employing four men. Three years later the plant was incorporated as the American Hoist & Derrick Co. In 1886 the plant was moved to a more advantageous position in the city and continued to grow until its buildings covered an area equal to many city blocks, and branches were established in all of the larger cities of the country.

Under the management of Mr. Crosby the concern handled from its earliest history many large Government contracts and during the War devoted its full resources to war-time construction. In the building of the

Panama Canal the firm supplied considerable equipment and was awarded the contract for the largest traveling crane in the world, a fifty-ton jib crane for use by the Government in the ship yard at Balboa in the Canal Zone. This crane was constructed at a cost of $80,000, weighed 590,000 pounds without counter or balance weights and fifteen freight cars were required to transport it from St. Paul. Members of the A.S.M.E. St. Paul Section and the St. Paul Society of Civil Engineers visited the plant while the crane was under construction.

Mr. Crosby's inventive genius was a material factor in the development of many mechanical devices manufactured by the company. He was the inventor and patentee of the Crosby wire rope clip, the American log loader, the American railroad ditcher, the American locomotive crane and more than a score of other devices used in heavy carriages and the hoisting business.

Mr. Crosby was very much interested in the civic affairs of St. Paul, was an active member of the St. Paul Association and was identified with other civic development organizations. He was much devoted to his Alma Mater and at his death left $100,000 for a mechanical engineering building to bear his name. He became a member of our Society in 1911. He was also a member of the Engineers' Society of St. Paul and of several national foundry and machinists' organizations. He belonged to the Masonic Order and was a Knight Templar and a Shriner.

WILLIAM WYMAN CROSBY

William W. Crosby, member of the firm of Haven & Crosby, architects and engineers, Boston, Mass., died on March 19, 1922. Mr. Crosby was born on March 3, 1871, in Arlington, Mass., where he received his early education. He attended the Massachusetts Institute of Technology from which he was graduated in 1893 with the degree of B.S. in mechanical engineering.

For one year he served as an instructor in steam engineering in the Institute, then becoming mechanical engineer for Otis Allen's Sons, Lowell, Mass. For several years he was principal of the Lowell Textile School and was instrumental in bringing that school to a high technical standard. Contemporaneously he acted as consulting engineer in the textile industry. Later he became chief engineer of the Brighton Mills, Passaic, N. J., where he was especially interested in the development of tire fabrics.

For a number of years he was a member of the firm of F. W. Dean & Co., engineers and architects in Boston, Mass., resigning to found the firm of Haven and Crosby of which he was a member at the time of his death.

Mr. Crosby became a member of the Society in 1900. He was a member of the Boston City Club, the Engineers' Club of Boston and of many other organizations.

HENRY HAVELOCK CUMMINGS

Henry H. Cummings, president of the Cummings Machine Works, Boston, Mass., died on December 25, 1922. Mr. Cummings was born in February, 1858, in Worcester, Mass., where he received his early education. His subsequent education was attained through private instruction and study.

From 1876 to 1881 Mr. Cummings worked as an apprentice and journeyman machinist in the shops of Boston, Mass. At the end of that period he started in the machine business for himself. This developed into the Cummings Machine Works.

Mr. Cummings was the inventor of the Brown bag-filling machine which packs Government seeds at Washington, the Cummings torsion meter, averaging revolution counters, sub-target rifle, speed clocks and the engine log system. He became a member of the Society in 1916.

FRANZ JOHANN DENK

Franz J. Denk, president of the Denk Engineering Co., Pittsburgh, Pa., died at his home in Clifton Heights, Pa., on May 18, 1922. Mr. Denk was born on February 27, 1874, in Mainz, Germany, where he received his early education. Later he was graduated from the Technische Hochschule, Darmstadt, Germany, remaining until 1903 as an instructor in engineering.

In 1903 he came to the United States where he became connected with the Pittsburgh Valve, Foundry & Construction Co. and the Duquesne Steel Co., Pittsburgh. In 1915 he established his own business as consulting engineer, specializing in fuel saving for glass and steel mills.

Mr. Denk became a member of the Society in 1919. He was also a member of the American Ceramic Society, the Technical Society of Pittsburgh and of the Masonic Order. He was well known as a writer for technical magazines.

THOMAS CLARK DILL

Thomas Clark Dill, formerly president of the T. C. Dill Machine Co., Inc., Philadelphia, Pa., died on January 6, 1922, of heart disease. Mr. Dill was born on April 10, 1864, in Berlin, N. J., and was educated in the public schools there.

He served his apprenticeship in Rickards Machine Shop, manufacturers of fire engines for the city of Philadelphia. In 1889 he started in business for himself as general machinist and in 1898, while building coining presses, rolling mills, etc., for the U. S. Mint at Philadelphia, found that he needed a slotter to do this work. Not being able to purchase one in time to suit his needs, he designed and built the first Dill travelling head slotter.

Mr. Dill became a member of the Society in 1913. He was also a member of the Old Colony Club of Philadelphia.

GEORGE H. EWING

George H. Ewing, manager of the Pacific Northwest Inspection Bureau, Seattle, Wash., died on March 1, 1922. Mr. Ewing was born in Glasgow,

Scotland, in 1870. He was educated at the Glasgow and West of Scotland Technical College and for six years was connected with Alex Nicholson & Co., Glasgow, as draftsman. From 1900 to 1909 he was managing director for Ewing & Lawson, Ltd., boilermakers and engineers of Glasgow.

After a year's travel through India, China, Japan, Argentine and Australia, Mr. Ewing, in the latter part of 1910, went to British Columbia, where he became resident engineer for Smith, Kerry & Chase, working on the design and construction of a hydroelectric plant for the municipality of Revelstoke. In 1912 he became chief engineer for the Mt. Hood Railway Light & Power Co. at Portland, Ore. From 1913 to 1916 he was inspector for R. W. Hunt & Co. at Seattle, Wash., and Vancouver, B. C., and at the close of that period became general manager for the Pacific Northwest Inspection Bureau.

Mr. Ewing became a member of the Society in 1919. He was also a member of the Institute of Engineers and Shipbuilders in Scotland.

GORDON SANDS FALK

Gordon Sands Falk, mechanical engineer in the gear department of The Falk Co., Milwaukee, Wis., died on December 31, 1922. Mr. Falk was born in Milwaukee, in May, 1887. He received his early education in the schools of that city and later attended the University of Wisconsin from which he received the degree of B.S. in 1910.

Upon graduation he entered the employ of The Falk Co. as a designer in the kerosene-engine department. Two years later he left this firm to become a partner in the Milwaukee Patent Leather Co., then newly organized to tan and manufacture patent leather. In 1915 Mr. Falk sold his interest in the company and returned to The Falk Co. in charge of the costs of the herringbone-gear department. The following year he became designing, estimating and sales engineer for this branch of the work.

Mr. Falk became an associate member of the Society in 1919.

WALTER G. FRANZ

Walter G. Franz, a prominent consulting engineer of Cincinnati, died at Biddeford, Me., on August 12, 1922. Mr. Franz was born in St. Louis, Mo., in July, 1875. He was educated in the schools of that city and attended the University of Missouri from which he was graduated in 1899 with the degree of B.S. in mechanical engineering.

He served an apprenticeship with the American Arithmometer Co., the Terminal Railroad Association and the Missouri River Commission, all in St. Louis. In his earlier work he was an inspector with the Bell Telephone Co. and later was employed by the Struthers-Wells Co., Warren, Pa., and by the Excelsior Wooden Pipe Co., Lynchburg, Pa. In 1905 he went to Cincinnati and became associated with Gustave W. Drach, architect. Three years later he began his own very successful career, becoming consulting engineer for the University of Cincinnati, Miami University and Tuskegee Institute. He also handled the engineering work on many of the

larger hospitals, office buildings and industrial plants in Cincinnati and the vicinity.

Mr. Franz was young and comparatively unknown when he went to Cincinnati but made for himself in but few years a place as engineer of unquestioned judgment and unswerving honesty. Quiet and unobtrusive in manner he made a silent appeal that reached and held men from every walk of life. The remarkable gathering at the final simple services proved that a man of unusual qualities had passed away.

Mr. Franz became a member of our Society in 1907. He was also a member of the American Institute of Electrical Engineers and belonged to the University, Rotary, Country and Engineers' Clubs of Cincinnati and to the Chamber of Commerce. He took a great interest in the Community Chest, providing for three French children, and was an enthusiast in the Boy and Girl Scout movement.

JAMES HENDERSON GEER

James Henderson Geer, who, until his retirement in 1915, was superintendent of construction of the Cambria Steel Co., Johnstown, Pa., died on March 15, 1922. Mr. Geer was born in Blairsville, Pa., on August 2, 1843. He received his early education in Johnstown schools, leaving at the age of fourteen to enter the employ of the Cambria Iron Co. From that time until 1915 he served the same concern with but one break — the period he spent during the Civil War in the Union Army.

Mr. Geer worked as a pattern maker until 1871 when he entered the drafting department. In 1881 he was master mechanic of the company and three years later assistant to the chief engineer. He held this position until 1896 when he was promoted to the position of superintendent of the mechanical department. In 1904 he became superintendent of the Cambria Steel Co. (formerly the Cambria Iron Co.) and held this position until 1915 when he was retired as a pensioner of the company. In 1917 he retired from the pension list and severed all relations with the concern.

In *McClure's Magazine* in a summary of Mr. Geer's activities it is said "What Mr. Geer accomplished in his particular line (the manufacture of steel from iron) cannot be adequately told in words; but it certainly is not asserting too much to say that his name suggests a power in the steel trade, a power that to a large degree controlled and directed the workings of the company which he so faithfully served and his name stands today as representative of leading steel men of his day."

Mr. Geer was a life member of the Society, joining the organization in 1884.

A. W. GIBBS

Alfred Wolcott Gibbs, chief mechanical engineer of the Pennsylvania Railroad System, died on May 19, 1922. Mr. Gibbs was born at Fort Filmore, N. M., on October 27, 1856. He received his preparatory schooling at Rutgers College Grammar School and then studied for two years in Rutgers College, when he entered Stevens Institute of Technology, being graduated in 1878.

In March, 1879, Mr. Gibbs entered the service of the Pennsylvania Railroad Co. as a special apprentice at the Altoona Shops, continuing such service until June 1, 1881, when he became draftsman with the Richmond and Danville Railway. In 1886 he was appointed master mechanic of the Atlantic and Charlotte Division of that line and two years later became superintendent of motive power of the Central of Georgia Railway. When that position was abolished he was again appointed master mechanic on the Atlantic and Charlotte Division.

In 1893 Mr. Gibbs reëntered the service of the Pennsylvania Railroad Co. as assistant mechanical engineer, and nine years later was advanced to the position of superintendent of motive power of the Philadelphia, Wilmington & Baltimore Railroad Co. On January 1, 1913, Mr. Gibbs was appointed general superintendent of motive power of the Pennsylvania Railroad Co. In 1911 he assumed the duties of the newly created position of chief mechanical engineer of that line and under the reorganization in 1920 became chief mechanical engineer of the Pennsylvania system.

Mr. Gibbs became a member of The American Society of Mechanical Engineers in 1920. He was also a member of the Mechanical Division of the American Railway Association, of the American Engineering Standards Committee, the American Railway Engineering Association, the American Society of Naval Engineers, of the Board of Managers of the Franklin Institute, of the Board of Managers of the Philadelphia Institute and Free Library and Past-President of the American Society for Testing Materials.

HERMAN F. GIELE

Herman F. Giele, for the past ten years foundry superintendent of the Grabler Manufacturing Co., Cleveland, Ohio, died in Baltimore, Md., on April 17, 1922. Over fifty years of his life was spent in the study and advancement of malleable-iron practice.

Mr. Giele began work in a foundry in 1869 and worked for several years in that and other foundries in central and southern Ohio. At the Erie Malleable Iron Co. he collaborated with B. J. Walker in conducting a long series of experiments over a period of eighteen years on iron mixtures, melting and annealing and on problems connected with furnace construction.

In 1893 Mr. Giele became connected with the Gould Coupler Co., Depew, N. Y., and personally supervised the building of the furnaces and the purchasing and installing of equipment in what was at that time the largest malleable iron foundry in the country. Later he built up the operating organization. In 1900 he organized the Meadville Iron Co., Erie, Pa., and as vice-president and general manager constructed and operated the plant. He returned to Erie in 1906 to assume charge of the erection and operation of the malleable plant of the Morse Iron Works, retaining a directorship in the Meadville Iron Co. In 1912 he became associated with the Grabler Manufacturing Co. at Cleveland. During the War he developed at this plant a successful method of producing the difficult castings for the Mills hand grenade.

Mr. Giele became a member of the Society in 1913.

WILLIAM GLEASON

William Gleason, president and founder of the Gleason Works in Rochester, N. Y., died at his home on May 24, 1922, in his 87th year. He was born in Tipperary County, Ireland, and came to this country at the age of fifteen. He went to work in the machine shop of Asa R. Swift in Rochester, serving his apprenticeship there and in the shop of I. Angell & Sons. During the Civil War he was a workman in the Colt Armory in Hartford, Conn. In 1865 Mr. Gleason returned to Rochester and formed the nucleus of the Gleason Works. For a few years he was in partnership with John Connell and James S. Graham, but this partnership was dissolved in 1873. It was just at this time that he perfected the first practical bevel-gear planer which revolutionized the bevel-gear-cutting machine industry, and Mr. Gleason went with the Kidd Iron Works as superintendent. Two years later he took over the business.

Expansion of the business necessitated a larger plant in 1896. In 1910 there was a further enlargement, and the business has extended not only all over the United States but to foreign fields as well. Beginning with six employees in a one-room plant, the Gleason Works now covers more than twenty-five acres and employs 800 men. During the War the plant employed 1700 men in making ordnance supplies for the Government, and expanded its production capacity to three times its pre-war capacity.

William Gleason piloted his firm through many dangerous business periods. His interest was always primarily with his employees and customers, and oftentimes he lost sight of cost and profit in an endeavor to turn out high-grade tools. He kept his standards high, and it was this integrity which brought him through periods which sometimes wrecked his competitors. At the fiftieth anniversary of the founding of the plant, old employees bore witness to his fairness and honesty.

Up until the latter part of 1920 Mr. Gleason took an active part in the management of the firm. He leaves to carry on his work two sons and Kate Gleason, his daughter, who is one of the three women members of the Society. Mr. Gleason became a member of the A.S.M.E. in 1897. He was a member of the Automobile Club of Rochester and of the Oak Hill Country Club.

JAMES E. GREENSMITH

James E. Greensmith, president of the Boston Scale & Machine Co., Boston, Mass., died on March 8, 1922. Mr. Greensmith was born in Burton-on-Trent, England, where he was educated as a mechanical engineer. His early business experience was in India where he was engineer in charge of the machinery on a tea plantation.

He came to this country in the eighties to take charge of the Pond Machine Tool Co. who were at that time preparing for a new factory in New Jersey. Mr. Greensmith supervised the construction and equipment of this plant as well as the design and development of a new type of heavy gun turning and boring lathe, later installed at the Watervliet Arsenal.

His next position was with the Portland Co., Portland, Me., as superin-

tendent, where he remained but a short time when he became superintendent of the Mason Machine Works, Taunton, Mass., holding this position for a number of years. For the past six years Mr. Greensmith was president and general manager of the Boston Scale & Machine Co.

Mr. Greensmith became a member of the Society in 1892. He was a member of the Engineers' Club of Boston and belonged to the Masonic Order.

FLORENCE M. GRISWOLD

Florence M. Griswold, who was often spoken of as the "Dean of Fire Insurance Engineers," and who was responsible for the standardization of fire-hose couplings, died on April 25, 1922. Mr. Griswold was born in Hoboken, New Jersey, in November, 1843, and received his education in the public schools there and at Wittenburg College, Springfield, Ohio. He served in the Union forces during the Civil War.

At the close of the war he entered the insurance business under the supervision of his father, Jeremiah Griswold, who was the author of many publications on various phases of the insurance business. Until 1875 he was connected with several of the principal fire-insurance companies in various responsible capacities, when he became general inspector of the Home Insurance Co. of New York. Since that time he had particular charge of the special hazards and technical work conducted by this company throughout the whole field of its operation.

When Mr. Griswold entered the insurance business the system was admitted to be one of guessing as to hazards and rates. He began to study the situation in an attempt to reach the scientific principles underlying it.

He made himself familiar with the methods of all classes of manufacturing industries and the fire hazards incident to each. He assisted in the organization of many of the inspection bureaus and had an active hand in the formulation of a number of schedules for rating industrial plants.

His investigations naturally led him into the field of fire extinguishment. For many years he worked strenuously to secure universal standards for all classes of fire-fighting facilities and utilities. The National Fire Protection Association selected him to head a special committee to secure the adoption of a universal standard for hose and hydrant threads. Persistent efforts in this direction had failed many times in the past. Mr. Griswold was finally able to secure for his coupling the endorsement of many of the leading and most influential organizations of this country, and its use became general in all parts of the country. In 1917 it was approved and adopted by the United States Bureau of Standards as the "National Standard Hose Coupling and Hydrant Fitting" to be used for public fire service.

Mr. Griswold became a member of The American Society of Mechanical Engineers in 1914. He was also a member of the Grand Army of the Republic, the American and New England Water Works Associations, an associate member of the International Association of Fire Engineers, and an honorary life member of the National Fire Protection Association.

FREDERICK HANDEL HAYWARD

Frederick Handel Hayward, a member of the Society since 1889, died on May 8, 1922, at his home in London, England. Mr. Hayward was born on April 30, 1860, in Providence, R. I., where he was educated. He was for some years chief engineer for the Ball & Wood Engine Co., and later for the firm of Armington & Sims. Still later, in association with others, he established the business of Julian Scholl & Co., New York City, for the manufactuure of steam rollers.

During the whole of this period he was in close touch with the Standard Oil Co., actively engaged in the erection of pipe lines, pumping stations, etc. In 1897 he went to England as European agent for the Ludlow Valve Manufacturing Co., Snow Steam Pump Works and the Pennsylvania Tube Works. In 1900 Mr. Hayward established his own business under the title of F. H. Hayward & Co., petroleum engineers, London, and was still actively engaged in the direction of this business at the time of his death.

HENRY HESS

Mr. Henry Hess, Past Vice-President of The American Society of Mechanical Engineers, died at his home in Atlantic City on March 23, 1922. He had been in failing health for the past two years but had been so much improved that he had visited his office in Philadelphia several days before his death.

Henry Hess was born in Darmstadt, Germany, in March, 1864, and came to the United States when a small boy. His education was received in the New York schools, and was supplemented by several years of additional schooling in Germany. After his return to the United States he was employed at the Watervliet Arsenal, Troy, N. Y., and later at the Niles Tool Works, Hamilton, Ohio. While in the employ of the latter concern he was sent to Germany to erect the German Niles Tool Works at Oberschoeneweide, near Berlin. He remained to have charge of the operation of this plant for two years after its erection. Upon his return to the United States in 1902 he organized the Hess-Bright Manufacturing Co. In 1912 he sold his interests in this company and organized the Hess Steel Corporation of Baltimore, with which organization he was connected at the time of his death.

Mr. Hess became a member of The American Society of Mechanical Engineers in 1906. From 1911 to 1914 he served the Society as Manager, and from 1914 to 1916 as Vice-President. In 1915 he presented to the Society a gift of $2000, the income from which is given annually as the Junior and Student Prizes for the best technical papers by Junior and Student Members. Mr. Hess was past-president of both the Society of Automotive Engineers and of the Philadelphia Engineers' Club; he was a member of the American Institute of Mining Engineers, the American Society for Testing Materials, The American Iron and Steel Institute, the American Electrochemical Society, the American Academy of Political and Social Science, The Franklin Institute, the New York Engineers' Club, the Art Club of Philadelphia and the Economics Club.

During his lifetime Mr. Hess was a contributor to various technical publications. He was a special lecturer at Columbia University on subjects on which he was an authority. Of late years he had been greatly interested in color photography and had lectured on this subject before various organizations.

Mr. Hess is survived by his wife, two daughters and a son, Mr. H. Lawrence Hess of Philadelphia, a member of the Society.

WILLIAM HEWITT

William Hewitt, a former manager of the Society and for forty-eight years in the service of the Trenton Iron Co., died on August 2, 1922. Mr. Hewitt was born in Trenton, N. J., on October 26, 1853. His early education was obtained at the New Jersey State Model School. Upon his graduation from that school in 1868 he worked for about a year at roll-turning and drafting at the works of the Trenton Iron Co., now the American Bridge Co. In the fall of 1869 he entered Lehigh University. He left this institution at the end of his sophomore year to enter as a junior the Stevens Institute of Technology which had just been opened. He had but one classmate and the two constituted the class of '73. Because of illness, however, Mr. Hewitt was graduated with the class of '74.

Soon after graduation he entered the service of the Trenton Iron Co., first as paymaster and later as assistant to his father, Charles Hewitt, President and general manager, in the work of designing, remodeling and installing new machinery. Upon his father's death, Mr. Hewitt's brother, Abram S. Hewitt, was made president and he, himself, was made vice-president, also holding from 1879 to 1888 the office of managing director. At the end of that period he became engineer in the new department of the business, concerned with the designing and building of aerial tramways and other methods of haulage involving the use of wire rope. Upon his brother's death, the works were sold to the American Steel & Wire Co., and the operation was continued under the old name of the Trenton Iron Co. In January of 1913, the old name was finally dropped. During all of these changes Mr. Hewitt continued to serve as tramway engineer and estimator, resigning on June 30, 1922, after forty-eight years of service.

Mr. Hewitt was one of the very early members of our Society, joining the organization in 1880; from 1884 to 1887 he served as one of the board of managers. He belonged also to the former Engineering Association of the South and was a member of the Theta Xi and Tau Beta Pi fraternities. He was the author of The Application of Wire Rope to Transportation, Power Transmission, etc., and also contributed numerous articles to the technical press as well as papers to the professional societies.

HARRY DENARD HILDEBRAND

Harry D. Hildebrand, president of the Hope Engineering & Supply Co., Pittsburgh, Pa., died on April 8, 1922. Mr. Hildebrand was born in Baltimore, Md., in September, 1870, where he received his education.

He served his apprenticeship with the Hall Steam Pump Co., later

becoming chief draftsman and sales engineer. In 1913 he became president of the Hope Engineering & Supply Co., engaged in directing and designing natural-gas pumping stations, the making of gasoline, the manufacture of pipe lines, etc.

Mr. Hildebrand became a member of the Society in 1916.

HOWARD M. INGHAM

Howard M. Ingham, efficiency engineer with offices in New York City, died on January 16, 1922. He was born on April 14, 1877, in Philadelphia, Pa. He was educated at Sheffield Scientific School, Yale University, and received his degree in 1897. The following year he took a post-graduate course in mechanical engineering.

From 1898 to 1904 he served his apprenticeship and gained his shop experience at the Southwark Foundry of Philadelphia, thereafter becoming associated in business with M. W. Kellogg. From 1907 to 1912 he was with the Bliss-Griffith Construction Co., resigning to open his own office as efficiency engineer. During the War he served for two years as Lieutenant, senior grade, in the U. S. Navy, inspecting and expediting the manufacture of ordnance. From 1919 until the time of his death he again engaged in private practice, specializing in efficiency work.

Mr. Ingham became a member of the Society in 1907. He was a member of the American Legion and belonged to several clubs.

CHARLES MURDOCH JAMIESON

Charles M. Jamieson, production and development engineer, died on May 8, 1922, in Tampa, Fla. Mr. Jamieson was born on November 12, 1882, in Bridgeport, Conn. He was a graduate of Sheffield Scientific School, Yale University, Class of 1904.

From 1904 until the outbreak of the War he designed tools and special machinery for various concerns about the country and as production engineer developed methods for increased production. During the War he had complete charge of the operation planning in the Wright-Martin airplane factories increasing the monthly production of Hispano-Suiza motors from 180 to 540 in the New Brunswick plant, when the lack of stock became the limiting factor. He was then transferred to the Long Island City Plant and did similar work there until after the Armistice. He then resumed his private practice as consulting and development engineer.

Mr. Jamieson became a member of the Society in 1919. He was also a member of several clubs in Bridgeport, Conn. and in Hamilton, Ohio.

BERNARD G. JEFFERIS

Bernard G. Jefferis of the Celluloid Co., Newark, N. J., died on January 21, 1922. Mr. Jefferis was born in West Chester, Pa., on October 1, 1896. He was graduated in 1920 from Pennsylvania State College with the degree of B.S. in industrial engineering.

His first employment was with Day & Zimmerman, Inc., Philadelphia,

Pa., as assistant to the appraisal engineer. After several months there he became connected with the Ace Motor Corporation, Philadelphia, as assistant to production engineer where he engaged in time study and planning. His next position was with the U. S. Cast Iron Pipe & Foundry Co., Burlington, N. J., as assistant to planning chief in charge of time-study work through the several plants. In March of 1921 he entered the task-setter production department of the Celluloid Co., where he was employed at the time of his death.

Mr. Jefferis became a junior member of the Society in 1921.

LAWRENCE BATES JENCKES

Lawrence Bates Jenckes, a director of the Crompton & Knowles Loom Works and a member of their Development Board, died at his home in Worcester, Mass., March 29, 1922. He was born in Millville, Mass., in March, 1867, and received the degree of Ph.B. from the Sheffield Scientific School, Yale University, in 1887.

Among his earlier works were the laying out of the present four-track line between New York and New Haven, and the surveying of the Croton watershed. For a time he was city engineer for Stamford, Conn., and later became superintendent of public works for the same city. In 1899–1900 he was assistant general manager of the Electric Light Axle & Power Co., New York City, leaving this position to enter the purchasing department of the Westinghouse Electric & Manufacturing Co. at Pittsburgh. For the past seventeen years, Mr. Jenckes had been connected with the Crompton & Knowles Loom Works, devoting much of his time to the invention and perfection of many types of looms, upon which subject he was regarded as an authority.

Besides membership in The American Society of Mechanical Engineers, dating from 1912, Mr. Jenckes was a member of the American Society of Civil Engineers, the Yale Club of New York, the Tatunek Country Club and the Squantum Association of Providence, R. I.

FRANK N. JEWETT

Frank N. Jewett, formerly sales manager of the Wagner Electric Manufacturing Co., St. Louis, Mo., died on January 4, 1922. Mr. Jewett was born on June 11, 1870, in Tioga, N. Y. He was graduated in 1893 from Cornell University. For two years after graduation he was engaged in experimental work in the shops of the Payne Engine Co., Elmira, N. Y., at the close of that period becoming their New York City salesman.

From 1896 to 1906 he was connected with Evans, Almirall & Co., first as salesman in New York City, and later as the western manager of their Chicago office. In 1906 he became Chicago district manager for the Wagner Electric Manufacturing Co. of St. Louis, and in 1909, sales manager of their St. Louis office. In 1913 Mr. Jewett was forced to resign from this position because of ill health but he remained with the company until the time of his death.

Mr. Jewett became a member of the Society in 1908. He was also

a member of the Society of Heating and Ventilating Engineers and an associate member of the American Institute of Electrical Engineers.

WILLS JOHNSON

Wills Johnson, chairman of the appropriation committee and assistant to vice-president in charge of operation of the General Motors Corporation, died suddenly on December 25, 1922, at his home near Greenwood, Va. Mr. Johnson was born on August 16, 1876, in Louisa County, Va.

Upon his graduation he took post-graduate work in mechanics and at the same time instructed in shop work. In 1897 he became instructor and assistant professor of mechanics at Clemson Agricultural and Mechanical College, S. C. During the summers he worked in the shops of the Richmond Locomotive Works, the drawing room of the Phoenix Bridge Co., Phoenixville, Pa., and the shops of the Straight-Line Engine Works, Syracuse, N. Y.

In 1903 he entered the University of Virginia and in 1906 was graduated with the degree of M.E., then becoming connected with E. I. Du Pont de Nemours & Co. as engineer in the construction of smokeless powder plants. From 1916 to 1920 he was in charge of the design division of the engineering department and did important work in directing the design of many of the larger War plants built by the company for Government work. In the fall of 1920 he joined the General Motors Corporation.

Mr. Johnson became a member of the Society in 1921. He belonged to a number of social clubs and organizations and was a member of the Delta Tau Delta fraternity.

THOMAS FRANCIS JUDGE

Thomas Francis Judge, general mill superintendent of the Anglo-Newfoundland Development Co., Ltd., Grand Falls, Newfoundland, died on May 12, 1922. Mr. Judge was born on December 19, 1877. He was educated in the Biddeford, Me., high school and the University of Maine, from which he was graduated in 1900 as an electrical engineer.

Upon graduation he spent one year in the drafting department of the Saco & Pette Machine Shops, of Biddeford, Me., then becoming assistant erecting electrical engineer for the Great Northern Paper Co., Millinocket, Me. His next position was with the Union Paper & Bag Co., Hudson Falls, N. Y., where he was engaged in the construction and operation of a hydro station. For two years he served as erecting electrical engineer with the Lawrentide Paper Co., resigning to become consulting and erecting engineer with George F. Hardy, in charge of work for the St. Croix Paper Co., the Champion Fibre Co., the Champion Paper Co., the A. N. D. Co, Ltd., and A. E. Reed & Co. During these years he also designed a great number of electrical plants for paper mills throughout the United States and Canada. From 1910 to 1913, Mr. Judge was chief electrician for the A. N. D. Co., Ltd., Grand Falls, Newfoundland. In 1913 he was appointed general superintendent of the Anglo-Development Co., Ltd., which position he was holding at the time of his death.

Mr. Judge was a member of the American Institute of Electrical

Engineers, the American Association for the Advancement of Science and of several other scientific associations. He became a member of our Society in 1921.

WILLIAM KANE

William Kane, president of the William Kane Manufacturing Co., Philadelphia, Pa., died on May 22, 1922. Mr. Kane was born on June 17, 1849, in Ireland.

In early life he developed and patented the first dry-joint automatic fire extinguisher which was later purchased by the General Fire Extinguisher Co. He developed and manufactured the first circulating gas water heater in the United States. In 1896 he patented his automatic steam boiler, gas fuel.

Mr. Kane, although seventy-two years of age, was actively engaged in directing the affairs of his company up to within a few days of his death. He became a member of the Society in 1917.

PIERRE O. KEILHOLTZ

Pierre O. Keilholtz, consulting engineer, Baltimore, Md., died January 21, 1922. Mr. Keilholtz was born in Baltimore, Md., on April 22, 1862. In 1884 he was graduated from the United States Naval Academy and for the following two years he was a graduate student at Johns Hopkins University.

In 1887 he became associated with the Electric Light, Power & Street Railway Corporation of Baltimore and was in responsible charge of the design, construction and operation work for the company until 1907 when he started private practice as a consulting engineer, in which field of work he was engaged at the time of his death.

Mr. Keilholtz became a member of the Society in 1901. He was also a member of the American Institute of Mining and Metallurgical Engineers, the American Institute of Electrical Engineers, the American Institute of Naval Architects and Marine Engineers and the American Society for the Advancement of Science. During the War Mr. Keilholtz served as a recruiting officer in the Aviation Section of the Signal Corps, U. S. Army.

O. W. KELLY

O. W. Kelly, president of the O. S. Kelly Co., Springfield, Ohio, and a member of the Society since 1885, died on October 31, 1922, at the age of seventy-one. He was educated abroad, attending the Polytechnic Institutes at Zurich, Switzerland, and Aix la Chapelle, Prussia.

He returned to this country in 1873 and entered the employ of Whiteley, Fassler & Kelly in Springfield, Ohio, where he spent three years at practical work in the shop and drafting room. In 1876 he became connected with the Champion Malleable Iron Works, working in the pattern room. The following year he was appointed superintendent of the works, which position he held until 1880 when he resigned because of poor health and went to Colorado. At Bonanza he erected a smelter for lead ores which he

operated successfully for nine months. In 1882 he returned to Ohio and became superintendent of the works of the Springfield Engine & Thresher Co., where he was located for a number of years. At the time of his death he was president of the O. S. Kelly Co.

JOHN T. KENYON

John T. Kenyon, formerly Major, Ordance Department, U. S. Army, died suddenly on March 1, 1922. Major Kenyon was born in New York City on January 23, 1856. His family moved to Springfield, Mass., when he was still young and he was educated in the schools of that city.

For a short while he worked in the shops of the Boston & Albany Railroad and then entered the employ of the Government in the Springfield Armory in 1880. In 1904 he was transferred to Rock Island Arsenal, Rock Island, Ill., to take charge of the small-arms department. When the Remington Arms Co. opened its Eddystone, Pa., plant for the manufacture of rifles, Major Kenyon accepted the position of consulting engineer to the works manager. Shortly after this, the United States entered the War and he was commissioned a major in the Ordnance Department of the Army, serving as an inspector of small arms. At the close of the War he was transferred to Rock Island Arsenal, soon afterward being discharged from the service, but remaining as a civilian in Government employ. In January, 1921, at the age of sixty-five, he was retired from active service.

Major Kenyon became a member of the Society in 1917. He belonged to the Army Ordnance Association and was a member of several fraternal organizations, including the Odd Fellows and the Masons.

ERNEST KRETSCHMER

Ernest Kretschmer, machine engineer with the E. W. Bliss Co., Brooklyn, N. Y., died on June 8, 1922. Mr Kretschmer was born on September 6, 1880, in Kolmar, Germany. He received his early education in the Royal Gymnasium of Schneidemull, and later was graduated from the Polytechnic Institute of Munich.

He obtained his practical experience in machine factories in Breslau and in 1905 came to this country where he became associated with the E. W. Bliss Co. He was with this concern for seventeen years. Mr. Kretschmer became a member of the Society in 1922.

ALBERT H. LA CASSE

Albert H. La Casse, superintendent and engineer, Sampson Magnesite Co., Mendota, Cal., died on February 10, 1922. Mr. La Casse was born on February 19, 1873, in Oswego, N. Y. He received his early education in the schools of Auburn, N. Y. Later he studied in evening schools to acquire his technical knowledge.

Mr. La Casse was first employed in 1890 by Connell & Dengler, manufacturers of wood-making machinery in Rochester, N. Y. He was next associated with Knowlton & Beach, of the same city, as a designer of

general paper-box manufacturing machinery. From 1897 to 1904 Mr. La Casse was connected for varying periods with the Hardy Cycle Co., Cobb & Hesselmeyer, the Risdon Iron Works and the L. C. Smith & Brothers Typewriter Co. In 1905 Mr. La Casse opened his own offices in Los Angeles where he designed machines for printing, folding and wiring ribbon on bottles; he was engaged in this business until 1914 when he became engineer and chief draftsman for the California & Hawaii Sugar Refining Co. In 1919 he was associated as engineer with the Meese & Gottfried Co., in San Francisco. He had been with the Sampson Magnesite Co. for a little over a year at the time of his death.

Mr. La Casse became a member of the Society in 1920. He was also a member of the Masonic Order.

JOHN ADAMS B. LANDELL

John A. B. Landell, of the Hanson-Van Winkle Co., Newark, N. J., died on June 3, 1922. Mr. Landell was born on February 27, 1888, in Philadelphia, Pa. He was educated in the Germantown Academy and the University of Pennsylvania, being graduated with the class of 1909, with the degree of B.S. in M.E.

Upon graduation he was employed by the Trinidad Lake Petroleum Co., Ltd., in the designing and erecting of a large topping plant for the recovery of gasoline from crude oil. He was with this firm for four years when he established his own business as consulting engineer on the design and construction of pipe-line pumping stations, gasoline extraction plants, etc. From 1917 to 1919 he served in the U. S. Army, gaining his pilot's license as an instructor.

When he was discharged from the Service, Mr. Landell became sales engineer for the Texas Gulf Sulphur Co., New York City. In 1920 he resigned from this position to become chief engineer of the Gasoline Recovery Corporation, also in New York City. At the time of his death he was associated with the Hanson-Van Winkle Co.

Mr. Landell became an associate member of the Society in 1921. He was a member of the University of Pennsylvania Club.

JAMES LANG

James Lang, estimating and contracting engineer of the John Inglis Co., Ltd., Toronto, Canada, died on June 5, 1922. Mr. Lang was born in Birkenhead, England, on April 2, 1858. He was educated at Watson's College, Edinburgh, and Edinburgh University. He served his apprenticeship with the W. G. Armstrong Co., of Manchester, where for a number of years he held the position of chief draftsman.

In 1888 Mr. Lang went to Canada where he became connected with the John Inglis Co., Ltd., Toronto, as draftsman and mechanical engineer. In 1897 he resigned to become chief engineer of the Northey Co., also of Toronto, remaining with that firm for four years when he opened offices as a consulting engineer. In 1904 he became chief mechanical engineer for the Canada Foundry Co., Toronto. In 1908 he returned to the John

Inglis Co., as estimating and contracting engineer which position he held at the time of his death.

Mr. Lang became a member of the Society in 1918.

CHARLES LANGLOTZ

Charles Langlotz, former mechanical superintendent of the Brooklyn Cooperage Co., Brooklyn, N. Y., died on September 29, 1922. Mr. Langlotz was born on March 15, 1862, in New York City, where he obtained his early education. He served his apprenticeship as a machinist and about 1880 operated the first power plant used for outdoor electric arc illumination in New York City.

In 1882 he entered the employ of the Brooklyn Cooperage Co. as assistant engineer, and shortly afterward, resolving to complete his technical education, enrolled in the evening course at Cooper Union from which he was graduated as a mechanical engineer in 1893. Under his management the present cooperage plants of the company in Brooklyn, Boston and New Orleans were constructed, and during this period he developed the first machine for automatically welding steel barrel hoops electrically.

In 1898 the construction of a large stave and heading mill was started under his direction at Poplar Bluff, Mo. Upon completion of the construction work, Mr. Langlotz was placed in charge of operation and remained at this post until 1908 when he returned to New York to design new plants required in the East. His last work was the construction of the plant at Georgetown, S. C. He retired from active service in 1921.

Mr. Langlotz became a member of the Society in 1894. He was also a member of the Brooklyn Engineers' Club.

SCHUYLER LAWRENCE

Schuyler Lawrence, long prominent in mining and railroad circles of Northern Mexico, died on January 9, 1922. Mr. Lawrence was born in Topeka, Kansas, in 1864. At the age of eighteen he went to Mexico where he was employed in the construction of the Mexican Central Railway from El Paso to Mexico City.

Upon the completion of the road he turned his attention to mining, settling in the City of Chihuahua, his principal operations being on properties in the vicinity of the camp of Santa Eulalia. At the beginning of the Mexican revolutions he left Mexico and returned to the United States, settling in Wysox, Pa., where he entered the apple-raising business.

During the War Mr. Lawrence was in charge of the Intelligence Bureau of the Mexican Division of the War Trade Board where his work was of unusually high character, drawing much favorable comment.

Mr. Lawrence became a member of the Society in 1902. He was also a member of the American Society of Mining and Metallurgical Engineers. He belonged to the Chihuahua Foreign Club and was the first president of the American Society of the State of Chihuahua.

RALPH A. LEE

Ralph A. Lee, specialty engineer, the Barrett Manufacturing Co., New York City, died on March 3, 1922. Mr. Lee was born in June 1884, in New York City, and was educated in Cooper Union Institute where he received the degree of B.S. in 1905 and M.E. in 1908.

From 1900 to 1909 he was associated with the following firms: National Meter Co., engineering department; H. R. Worthington, pump draftsman; Power Specialty Co., draftsman on superheaters; Barrett Manufacturing Co., mill construction; U.S. Wood Preserving Co., engineer in charge of construction; Walter Kidde Co., engineer. In 1909 he again became connected with the Barrett Manufacturing Co., as draftsman, and the following year was advanced to the position of chief draftsman; three years later he became assistant to the assistant general manufacturing manager and still later mechanical engineer. At the time of his death he was the specialty engineer for the company.

Mr. Lee became a junior member of the Society in 1909 and was promoted to the grade of associate member in 1914.

HERBERT K. MCBRIDE

Herbert K. McBride, structural and mechanical engineer, Wilson & Co., Chicago, Ill., died on January 7, 1922. Mr. McBride was born on September 14, 1870, in Olathe, Kansas, where he received his early education. Later he supplemented this with courses in the International Correspondence School.

From 1893 to 1898 he was connected with the Consolidated Electric Light & Power Co., Kansas City, resigning at the close of that period to take a position with Swift & Co., also of that city. The following year, 1899, he became chief electrician at the Kansas City plant of Wilson & Co. (then Schwarzschild & Sulzberger); in 1906 he was advanced to the position of chief engineer of the same plant and six years later became superintendent of all of the company's power houses, with headquarters in Chicago. At the time of his death, he was structural and mechanical engineer for the same concern.

Mr. McBride became a member of the Society in 1918.

EMERSON MCMILLIN

Emerson McMillin, head of the banking house of Emerson McMillin & Co., New York City, chairman of the Board of the American Light & Traction Co., and president of a dozen other lighting and traction companies throughout the country, died on May 31, 1922. Mr. McMillin was born in Ewington, Ohio, on April 16, 1844. He attended a country school until he was about ten years old when he was compelled to go to work. He subsequently gained his knowledge of science, art and literature by reading and studying after his day's work was done.

At the age of seventeen he responded to Lincoln's call for men and enlisted in the 18th Ohio Infantry as a private, serving his country until the

middle of 1865 when he was honorably discharged with the rank of second lieutenant. He had been wounded five times. At the close of the war he went into merchandising but soon sought more congenial employment in a gas works and for fifteen years he devoted all his leisure time to the study of civil and mechanical engineering, chemistry and geology. In 1883 Mr. McMillin ceased to give further attention to the iron and steel industry in which he had been interested and devoted himself entirely to the gas business, acquiring control of small works by purchase or lease throughout the West. In 1889 he gradually withdrew from the personal management of the gas works and took up almost exclusively the financial phase of the business and through banking houses in New York finally controlled several large corporations. In 1891 he established, in partnership with Col. Henry B. Wilson, the banking house of Emerson McMillin & Co.

During his whole life Mr. McMillin took a keen interest in civic matters, and was for many years actively interested in the National Civic Federation. He was appointed a member of the "Committee of One Hundred" which had for its object the establishment of a "True International Court of Justice." He was one of the speakers at the World Court Congress in 1915 in Cleveland, Ohio. He was also one of the principal organizers of the World Court League. He served on the board of governors of many institutions.

When the War was declared in 1914 and before this country became involved Mr. McMillin was one of the largest contributors to the aid of Italy, and in recognition of his interest received from King Victor Immanuel an honor medal.

He belonged to so many different clubs and organizations that it is hardly possible to name them all, but among them may be mentioned the Franklin Institute, the American Gas Association, the Natural Gas Association, the Academy of Political Science, the Numismatic Society of America, the Union League, the Bankers, Lawyers, Engineers and Chemists Clubs, the Army and Navy Club, the National Civic Federation, the Merchants' Association of New York and the Aero Club of America. Mr. McMillin became a life member of our Society in 1895.

JOHN G. MAIR-RUMLEY

John G. Mair-Rumley, formerly director of Messrs. Gwynnes, Ltd., Hammersmith Iron Works, London, and connected also with P. J. Mitchell Shaw & Co., Ltd., London, died early in 1922. Mr. Mair-Rumley was born on September 5, 1844. He was educated at Rugby and served his apprenticeship with Easton & Amos, mechanical engineers. He was associated with the North Eastern Railway and with Simpson & Co., before becoming connected with the Hammersmith Iron Works.

Mr. Mair-Rumley was a life member of the Society, joining the organization in 1892. He was also a member of the Institution of Mechanical Engineers and of the Institution of Civil Engineers in Great Britain. He was awarded the Watt Medal and three Oxford premiums for various papers presented before the latter society.

LESTER MANGELS

Lester Mangels, formerly assistant engineer with Murrie & Co., Inc., New York, N. Y., died on May 28, 1922. Mr. Mangels was born in Brooklyn, N. Y., on March 4, 1899. He was educated in the Brooklyn schools and Cooper Institute where he took the four-year course in architectural construction.

Upon graduation he started work with the Scottish Mexican Oil Co., New York, N. Y., as draftsman. A short while later he became connected with the Empire City Subway Co., New York City, also as draftsman. In April, 1917, he was employed by Westinghouse, Church, Kerr & Co., New York, N. Y., where he was located for about six months when he became assistant engineer with W. C. K. & Co. During the War he was in Muscle Shoals, Ala., as assistant engineer for that company on Nitro Plant No. 2. He remained with the firm until 1920 when he became squad chief with Dwight P. Robinson & Co., New York, N. Y., in charge of the design and preparation of plans for power and industrial plants. In January, 1921, he became associated with Murrie & Co., Inc., as assistant engineer in charge of the mechanical equipment appraisal of the N. Y. Edison power plants. He resigned from this position because of illness.

Mr. Mangels became a junior member of the Society in 1921.

EARL POTTER MASON

Earl P. Mason, production engineer with the Anti-Corrosion Engineering Co., New York City, died on August 14, 1922. Mr. Mason was born on May 18, 1876, in Providence, R. I., where he received his early education. Later he attended the Massachusetts Institute of Technology, being graduated with the class of 1897.

Upon graduation he entered the employ of the Eastman Kodak Co., Rochester, N. Y., where he remained a year, then becoming superintendent of the experimental department of the Draper Co., textile manufacturers in Hopedale, Mass. From 1902 to 1914 Mr. Mason was associated with the Newport Engineering Works, Newport, R. I., and as secretary and superintendent was responsible for the expansion of that concern from a small to an extensive plant.

In 1917 he resigned to accept a commission as lieutenant in the U. S. Naval Reserve Force. Early in 1919 he was promoted to the rank of lieutenant commander and a little later became a commander. In December of that year he was relieved from active duty at his own request and immediately took over for the U. S. Shipping Board the supervision of the maintenance and repair work in the New York District. He resigned from this position in May 1920 to become connected with the Anti-Corrosion Engineering Co.

Mr. Mason became a member of the Society in 1914. He was also a member of the Society of Naval Architects and Marine Engineers and belonged to a number of clubs and fraternal organizations, including the Masonic Order.

STANLEY G. MILLER

Stanley G. Miller, vice-president and general manager of the Crex Carpet Co., St. Paul, Minn., died on March 18, 1922. Mr. Miller was born in Philadelphia, Pa., on March 6, 1867. He was associated with the textile industry practically all of his life. After being graduated from the Philadelphia Textile School he was for many years connected with the Southern Cotton & Oil Co., Charlottesville, Va.

In February of 1910 he became vice-president and general manager of the Crex Carpet Co. in St. Paul and was soon a rcognized leader in the business life of the community. He did very unusual work for his firm by improving the machinery and standardizing the product and by his interest in the establishment of right relations between employer and employee. Due to his work in this direction he was elected a director of the St. Paul Association of Public and Business Affairs and in consequence was made chairman of the Sub-Division of Industrial Relations. Shortly afterward he became president of the St. Paul division of the National Safety Council to which he devoted his time and energy unsparingly.

Mr. Miller became an associate of the Society in 1917 and since the summer of 1921 was chairman of the St. Paul Section.

MARK RICHARDS MUCKLE, JR.

Mark R. Muckle, Jr., for the last twenty years head of the engineering firm of M. R. Muckle, Jr., & Co., Philadelphia, Pa., died on May 19, 1922. Mr. Muckle was born on March 3, 1857, in Philadelphia, Pa., and was educated in private schools of that city.

He served his apprenticeship with Wm. Sellers & Co., machine-tool builders in Philadelphia, and afterwards became a draftsman with that company. In 1883 he became a member of the firm of Ferrell & Muckle Enterprise Hydraulic Works, designers and builders of pumps and steam and hydraulic machinery. Later he became head of the firm of M. R. Muckle, Jr. & Co., consulting and contracting engineeers for plants operated by steam, electricity and gas.

Mr. Muckle was one of the early members of the Society, having joined in 1885. He was one of the founders of the Engineers' Club of Philadelphia and one of the original members of the Society of Naval Architects and Marine Engineers.

WILLIAM T. MURPHY

William T. Murphy, general manager of the Standard Machinery Co., Cranston, R. I., died on July 9, 1922. Mr. Murphy was born on June 1, 1882, in Providence, R. I., and was educated in the public schools of that city and Brown University from which he was graduated in 1903 with the degree of M. E.

He was first employed by the Taft-Pierce Manufacturing Co., Woonsocket, R. I., in their machine shop, where he remained for about a year,

then becoming connected with the Gorham Co. as a designing draftsman in the mechanical department. Upon the completion of this apprenticeship he accepted the position of tool designer with the American Locomotive Co. where he was located for about two years, when he became mechanical engineer for the Standard Machinery Co. He was associated from that time, 1907, until the time of his death with this concern.

Mr. Murphy became a member of the Society in 1911. He was also a member of the Automotive Engineers of Providence and of a number of clubs.

SAM L. NAPHTALY

Sam L. Naphtaly, of the Great Western Power Co. of California, San Francisco, Cal., died on June 25, 1922. Mr. Naphtaly was born in November, 1874, in San Francisco, where he received his early education. Later he was graduated from the University of California.

He was first employed by the California Light & Power Co., where he remained a year when he took a position with the Central Light & Power Co., installing a 1200-kw. plant. From 1900 to 1905 he was with the S. F. Gas & Electric Co., in charge of construction and engineering. In 1905 this company was merged with the Pacific Gas & Electric Co. and Mr. Naphtaly was made engineer of the latter concern. Later he became general manager of the Reno Light, Power & Water Co., and then general superintendent of the City Electric Co. of San Francisco. He was for a short period president of the Oakland and Antioch Railway and vice-president and general manager of the Los Angeles Shipbuilding and Dry Dock Co. Early in 1922 he entered the employ of the Great Western Power Co.

Mr. Naphtaly became a member of the Society in 1907.

HENRY NOYES

Henry Noyes, of the firm of Noyes Brothers, Sydney and Melbourne, Australia, died on March 20, 1922. Mr. Noyes was born in England in 1861 and received his education in English schools. In 1888 he and his brother Edward left England for Australia where they established their own business in Sydney and later in Melbourne. The firm held a number of important agencies, British, American and others, and carried out many large engineering works, including the installation or initiation of electric tramways at Dunedin, N. Z., Freemantle and Adelaide.

Mr. Noyes was a recognized authority on steel and a member of the Iron and Steel Institute of Great Britain and the American Institute of Mining and Metallurgical Engineers. During the War he took an active part in recruiting, paying a special visit to England to consult Sir John Monash in reference to that work. He gave large gifts to the Red Cross and provided a complete X-ray outfit for use in hospital work.

Mr. Noyes was a life member of our Society, joining the organization in 1899.

HENRY B. PRATHER

Henry B. Prather, consulting engineer, Cleveland, Ohio, died on March 9, 1922. Mr. Prather was born on September 14, 1867, in Indianapolis, Ind., and spent his early life in that vicinity. He received his engineering education in Cornell University, being graduated in 1890.

Until 1897 Mr. Prather was employed as chief draftsman for the Buffalo Forge Co., Buffalo, N. Y. From that time until 1900 he was the district representative for the same company in Pittsburgh and Cleveland. In 1901 he became associated with the American Stoker Co. and the following year he was with the Charles Bradley Co. Toward the latter part of 1902 the Reisinger-Prather Co. was formed, with headquarters in Pittsburgh, doing business as contracting engineers; three years later Mr. Prather founded the Prather Engineering Co., contracting engineers, with headquarters in Cleveland, and until 1907 he held the office of president in the firm. In that year Mr. Prather opened an office for himself as consulting engineer, continuing in this capacity until his death.

Mr. Prather became a member of the Society in 1893. He was also affiliated with the Cleveland Engineering Society, the Cleveland Chamber of Commerce, the Technical Association of the Paper & Pulp Industry, the Knights Templar and the Detroit Athletic Club.

ARTHUR BREESE PROAL

Arthur Breese Proal, for twenty-three years connected with the Robins Conveying Belt Co., New York City, died on August 21, 1922. Mr. Proal was born in New York City in 1869 and received his early education in the city schools, later attending Pratt Institute and for a period Stevens Institute of Technology.

For one year he served as assistant on an ocean liner, then becoming associated for eight months with the Metropolitan Telegraph & Telephone Co. At the end of that period he became constructing and operating engineer with the Newark Electric Light & Power Co., Newark, N. J., where he was located for five years. His next position was with the New Haven Street Railway Co., as superintendent, from which he resigned to become chief engineer of the New York & Staten Island Electric Co. and the New York & New Jersey Ferry Co. Two years later he became associated with the Robins Conveying Belt Co.

Very little had been done with conveying belt machinery up to 1899 and Mr. Proal was credited with recognizing the possibilities in this method of handling raw material. He was engaged in the development of the Panama Canal and the Baltimore & Ohio Railroad coal-handling plants at Baltimore. In the latter 9000 tons of coal are handled in one hour His last work, which has just been completed, was the coal-handling plant at Astoria, L I., consisting of three bridges and an extensive conveying machinery layout, involving many new features.

Mr. Proal became a member of the Society in 1905. He also belonged to the American Society of Civil Engineers.

WILLIAM E. PULIS

William E. Pulis, president and treasurer of the Wilkata Folding Box Co., Newark, N. J., died on November 3, 1922. Mr. Pulis was born on March 29, 1883, in Paterson, N. J., and received his early education in the Rogers Military Academy of that city, later attending the Paterson Classical and Scientific School. He served his apprenticeship in the machine shop of the Robins Conveying Belt Co., Passaic, N. J., then becoming assistant shop engineer of the Portland Shipbuilding Co., Portland, Me.

In 1905 Mr. Pulis became associated with H. L. Gantt, working under his direction first at the Chester Steel Castings Co., Chester, Pa., and then at the Sayles Bleacheries, Saylesville, R. I. In 1909 he was made superintendent of bleachery No. 3, and in the period which followed he directed the reconstruction and rebuilding of the bleaching department. From 1912 to 1916 he was Mr. Gantt's representative at the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa., and in 1917 represented him at the Celluloid Co., Newark, N. J. Soon after he severed his connection with Mr. Gantt to become works manager, vice-president and member of the Board of Directors of the same concern, which positions he held until 1921 when he resigned because of failing health. In the spring of 1922 he organized and became president and treasurer of the Wilkata Folding Box Co.

During the War Mr. Pulis served in an advisory capacity in the Ordnance Department of the Army and also with the Emergency Fleet Corporation. He became a junior member of the Society in 1909 and was advanced to full membership in 1916. He belonged to a number of clubs in Newark.

ROBERT QUAYLE

Robert Quayle, who, until his retirement in the early part of 1922, was general superintendent of motive power of the Chicago & Northwestern Railway Co., died on September 13, 1922. Mr. Quayle was born in Douglas, Isle of Man, on November 23, 1853, and came to the United States in 1868, where he started work for the Chicago & Northwestern Railway as a machinist's apprentice.

In 1877 he was made assistant foreman of the Chicago shops and in 1880 foreman of the car and machine shops. In 1885 he was appointed general master mechanic of all the company's lines in Iowa. Six years later he became general master mechanic of the M. L. S. & W., which road was absorbed by the C. & N. W. in 1893. The following year he was appointed general superintendent of motive power for the C. & N. W., which position he held until his retirement in 1922, after fifty-four years of continuous service.

In 1891 Mr. Quayle built and installed on the M. L. S. & W. the first locomotive-testing plant ever used on a railway. One of his achievements was the designing and building of the first Atlantic type of locomotive in 1900. He also designed and developed the sloping tanks which solved the problem the locomotive fireman encountered in getting at the coal.

Mr. Quayle became a member of the Society in 1914. He was also a member of the Franklin Institute, the Western Society of Engineers and a past-president of the American Railway Master Mechanics' Association.

ALANSON D. QUINT

Alanson D. Quint, former manager of the Turret Drill Co., Hartford, Conn., died on July 16, 1920. Mr. Quint was born on March 27, 1847, in Lexington, Me., where he received his early education. He served his apprenticeship as a machinist in Whitman's Machine Shop in Winthrop, Me., and subsequently worked as a journeyman machinist and tool maker in different towns of Maine, New Hampshire and Massachusetts.

In 1870 he was engaged by Colt's Patent Fire Arms Manufacturing Co., Hartford, Conn., as a tool maker, later becoming master machinist and foreman with the company. In 1892 he resigned from this position to become manager of the Turret Drill Co., Hartford, with which firm he was associated until his retirement in 1919.

Mr. Quint was a life member of the Society, joining the organization in 1891. He belonged to the Masonic Order.

JOHN RAE

John Rae, secretary and treasurer of the MacDonald & Campbell Co., Philadelphia, Pa., died on January 11, 1922. Mr. Rae was born on August 13, 1871, in Ely, England. He was educated in the schools of Philadelphia and was a graduate of the Franklin Institute where he specialized in machine design.

He was first employed by Wm. Wharton, Jr., & Co. as a draftsman in railroad engineering. Subsequently he held the following positions all with firms in Philadelphia: The year of 1889–90 he spent with the Link-Belt Engine Co., working on conveying machinery. He then became connected with A. Whitney & Sons, manufacturers of car wheels. For four years he was with the firm of William Wood & Co. as draftsman, assisting in the organization of a department system, the rearranging of machinery, etc. In 1898 he became associated with Wm. Sellers & Co. as draftsman and examiner, remaining with that company until 1909 when he became general manager and director of the Kearns Manufacturing Co., Philadelphia. Later he held the position of service engineer with the Tabor Manufacturing Co. and was for a short period shop engineer with the Bement-Niles Works in Philadelphia, resigning from the latter position to become secretary and treasurer of the MacDonald & Campbell Co.

Mr. Rae became a member of the Society in 1912. He belonged to several clubs and organizations and to the Masonic Order.

LINGAN STROTHER RANDOLPH

Lingan Strother Randolph, consulting engineer and professor of mechanical engineering at the Virginia Polytechnic Institute for twenty-five years,

died at his home in Baltimore, Md., on March 7, 1922. Professor Randolph was born in Martinsburg, W. Va., on May 13, 1859. He was educated in the Shenandoah Valley Academy, the Virginia Military Institute and Stevens Institute of Technology, where in June, 1921, he was awarded the honorary degree of doctor of engineering.

He spent his apprenticeship in the shops of the Baltimore & Ohio Railroad, then becoming engineer of tests with the New York, Lake Erie and Western Railroad. Later he became superintendent of motive power for the Florida R. & N. Co. from which position he resigned to accept a similar situation with the Cumberland & Pennsylvania Railroad. He returned to the Baltimore Railroad as engineer of tests in the Mount Clair shops. In 1893 he became professor of mechanical engineering in the Virginia Polytechnic Institute, Blacksburg, Va., where he remained for twenty-five years, retiring as dean of the engineering school.

The War recalled him to active service and he joined the research section of the U. S. Shipping Board, Emergency Fleet Corporation, Philadelphia. Upon the completion of his work there he opened offices in Baltimore, Md., as consulting industrial engineer, in which activity he was engaged at the time of his death.

Professor Randolph was a very early member of the Society, joining the organization in 1884. He was also a member of the American Society of Civil Engineers, a Fellow of the American Institute of Electrical Engineers and a member of the International Association of Testing Materials. He belonged to the National Arts Club of New York and the Engineers' Club of Philadelphia.

RICHARD DURFEE REED

Richard D. Reed, after a long illness, died on August 16, 1922. Mr. Reed was born in Westfield, Mass., on February 4, 1862. He was educated in the public schools of Westfield and at the Massachusetts Institute of Technology.

His first professional experience was gained with the H. B. Smith Co., Westfield, where he engaged in iron molding and steam fitting. With the exception of two years, 1882 to 1884, when he was constructing and designing low-pressure steam and water-warming apparatus for E. H. Cook & Co. in Elmira and Rochester, N. Y., and Chicago, Ill., his entire business career was spent with the H. B. Smith Co., where he constructed and designed cast-iron boilers and radiators for steam and water warming.

Mr. Reed served a two-year period on the councils of both the National Metal Trades Association and the National Founders Association. For two years previous to his becoming a member of the Society in 1915, he assisted in the work of the Boiler Code Committee and in 1915 was appointed a member of this Committee in whose work he took a deep interest, giving as active support as his health would permit up to the time of his death.

RICHARD H. RICE

Richard H. Rice, late manager of the Lynn Works of the General Electric Co., an outstanding member of the engineering profession and a leading authority on steam turbines, died on February 10, 1922, at Bolton on Lake George, N. Y. Mr. Rice was born on January 9, 1863, at Rockland, Me. He was educated in public schools and Stevens Institute of Technology from which he received his M.E. degree.

He served his apprenticeship with the Pittsburgh, Columbus, Cincinnati and St. Louis Railroad, then becoming a draftsman at the Bath Iron Works in Maine. His enthusiasm and ability as chief draftsman with E. D. Leavitt, Cambridge, Mass., brought him the position of general superintendent of the Wm. A. Harris Steam Engine Co., Providence, R. I. Later he was secretary-treasurer of the Rice Sargent Engine Co. and the Providence Engineering Works.

In 1903 he became associated with the General Electric Co. as engineer in the turbine department of their Lynn Works where he started work on the development of the turbine to its present extensive proportions. In 1918 he was made manager of the plant.

Mr. Rice combined the qualities of a skilled engineer and inventor with marked ability as an executive. He had a warmth of human sympathy, a regard for the feelings of others and a democratic spirit which made him a man easy of approach. He leaves as a monument to his memory the plan of representation in effect at the Lynn Works whereby his fellow employees reap the advantages of a new, yet very old idea — good will, friendship and coöperation.

Many business and professional organizations counted him as a member. He became a member of The American Society of Mechanical Engineers in 1890 and from 1904 to 1907 served as one of its managers. For the first two years of its existence he was president of the Associated Industries of Massachusetts; he was a past-president of the National Conference of State Manufacturers' Associations. He was also a member of the Providence Engineering Society and the American Institute of Electrical Engineers. He was a member of the Boston Chamber of Commerce, a director of the Lynn Chamber of Commerce, a member of the State Committee on Unemployment and during the War served as a member of the Lynn Fuel Commission. He was the author of several technical papers dealing with turbine engines and the inventor of some fifty patented devices for use of steam, air and water, chief among which was his design for the first turbo-blower for blast furnaces in America. In June of 1921 he was awarded degree of Doctor of Engineering by Stevens Institute.

CARL JOHN ROELKER

Carl J. Roelker, a member of the firm of Roelker & Lee, consulting engineers of Richmond, Va., died on July 7, 1922. Mr. Roelker was born in Osnabruck, Germany, in October, 1873. His education included two years study at Lehigh University and a year's training in the drafting room of

the Bureau of Steam Engineering, U. S. Navy. From 1897 to 1905 he was engaged in marine engineering with various companies, as follows: Newport News Shipbuilding and Dry Dock Co., in charge of the arrangement of the flooding system for the U.S.S. *Illinois;* Wm. R. Trigg Co., in charge of the design and arrangement of machinery and dock tests of the U.S. Destroyers *Dale* and *Decatur;* dock tests for the U.S. torpedo boats *Shubrick, Stockton* and *Thornton;* New York Shipbuilding Co., design of a triple-expansion engine for dredges and also the design of a hydraulic-launching gear; Standard Oil Co., design and arrangement of machinery for the S.S. *Capt. A. F. Lucas.* From 1901 to 1903 he served as an instructor in mechanical engineering at the Virginia Mechanics Institute in Richmond.

Mr. Roelker became a member of the firm of Roelker & Lee in 1905, with which he was associated at the time of his death. He became a member of the Society in 1909.

WILLIAM HENRY SAWTELL

William Henry Sawtell, sales engineer with the Mehl Machine Tool & Die Co, Roselle, N. J., died on December 16, 1922. Mr. Sawtell was born in April, 1870, in Worcester, Mass., where he received his early education, which he later supplemented with special technical correspondence courses.

His first employment was with the Waltham Watch Co., Waltham, Mass., where he was located for twelve years as machine-operating and assistant job master. From 1899 to 1904 he was connected with the firm of Alcorn & Ames, Waltham, Mass., as working foreman on tool and machine design and construction. The following two years he was foreman of the tool and machine department of the E. Howard Watch Co., also in Waltham, and at the end of that time became master mechanic with the L. S. Starrett Co., Athol, Mass. From 1911 to 1913 he was associated with the Geo. D. Day Machine Co., Laconia, N. H., as superintendent; B. C. Ames & Co., Waltham, as designer; Remington Tool & Machine Co., Woburn, Mass., as superintendent; American Watch Tool Co., Waltham, Mass., as designer.

From 1913 to 1915 he was superintendent of the Universal Trolley Wheel Co., Northampton, Mass., resigning to become mechanical engineer with the Athol Machine Co., Athol, Mass. From 1918 until 1921 he was associated with the Worcester Shock Absorber Co., Worcester, Mass., as superintendent. At the time of his death he was sales engineer for the Mehl Machine Tool & Die Co.

Mr. Sawtell became a member of the Society in 1918.

ARMAND J. SCHENKLE

Armand J. Schenkle, connected with the Reading Foundry & Supply Co., Reading, Pa., died on September 10, 1922. Mr. Schenkle was born on August 9, 1879, in Philadelphia, Pa., where he received his early education. Later he attended Drexel Institute from which he was graduated in 1911 with the degree of M.E. Upon graduation he was employed by the Philadelphia Iron Works as a designer, remaining there for a year when he became superintendent of the Natural Manufacturing & Engineering Co.

In 1916 Mr. Schenkle became connected with the Pennsylvania Iron Works, Eddystone, Pa., as a designer of tools and fixtures, very shortly being appointed general superintendent of the Works. He resigned from this position to become associated with the Baldwin Locomotive Works as chief inspector on artillery ammunition. In 1917 he entered the employ of the United States Government as an Army inspector of Ordnance. He rendered valuable service in organizing and equipping various plants in the Chicago district. He was associated with the Chicago district office until June of 1921. At the time of his death he was with the Reading Foundry & Supply Co.

Mr. Schenkle became an associate member of the Society in 1919 and was advanced to full membership in 1922. He was a member of the Masonic Order.

FREDERICK C. SCHOFIELD

Frederick C. Schofield, designing engineer with the United Cigarette Machine Co., Lynchburg, Va., died on September 26, 1922. Mr. Schofield was born on August 23, 1870, in Glenrock, Pa. He received his early education in the schools of Frostburg, Md., which he supplemented later with technical correspondence courses.

During his professional career he was associated with the following firms: National Electric Supply Co., Washington, D. C.; Cardwell Machine Co., Richmond, Va., as a machine designer; Tobacco Stemming Machine Co., Baltimore, Md., as designing engineer; R. J. Reynolds Tobacco Co., Winston-Salem, N. C., as designing engineer; Comas Cigarette Machine Co., Salem, Va., and the International Filler Corporation, Petersburg, Va., as designing engineer with both firms. In 1921 he became associated with the United Cigarette Machine Co., as designing engineer.

Mr. Schofield became a member of the Society in 1921.

JAMES B. SEAGER

James B. Seager, formerly general manager of the Seager Engine Works, Lansing, Mich., died on April 21, 1922. Mr. Seager was born in Lansing, Mich., on December 10, 1868. He was graduated from the Michigan Military Academy in 1886 and from the Michigan College of Mines in 1888. He then took a two-year post graduate course at the Massachusetts Institute of Technology.

From 1890 to 1894 he was in charge of the mine engineering for the Tylers Forks Mining Co., Plumer, Wis.; from 1894 to 1897 he was general manager of the Mackolite Fire Proof Construction Co., Chicago, Ill., on the erection and fireproofing of steel structures in Chicago and the middle West. For the next five years he was general manager of the Helvetia Copper Co., Helvetia, Ariz., in charge of the design and erection of smelting works and railroad, underground and surface workings of the mines. At the end of that time he became general manager of the Olds Gas Power Co., later the Seager Engine Works, Lansing, Mich.

Mr. Seager became a member of the Society in 1908.

COLEMAN SELLERS, JR.

Coleman Sellers, Jr., was born in Cincinnati, Ohio, September 5, 1852, and died after an acute illness of several months in Bryn Mawr, Pa., on August 15, 1922. Mr. Sellers' health became impaired as a result of his heavy duties during the War as head of the draft board in his district.

After an early education received in the private schools of Philadelphia, Mr. Sellers entered the University of Pennsylvania, where he was a first-honor man throughout his course. He was graduated in 1873 with the degree of Bachelor of Science, and three years later received his Master's degree for shop tests and a thesis relating to steam-boiler injectors. He entered the employ of William Sellers & Co. in November, 1873, and from 1886 to 1902 was assistant manager. In 1902 he was appointed engineer, and on the death of William Sellers was made president of the company. He continued in these capacities until the time of his death.

Mr. Sellers was of the sixth consecutive generation of a family engaged in the mechanical arts. His father, Coleman Sellers, a Past-President of The American Society of Mechanical Engineers, had a long career as an inventor. The sound judgment, ingenuity and experience of Coleman Sellers, Jr., contributed in full measure in maintaining the high reputation of his company as a designer of machine tools and appliances in its special field.

From 1890 to 1893 Mr. Sellers served The American Society of Mechanical Engineers on its Board of Managers; he was among the earliest to join the Society, having become a member in 1882. He was also a member of the American Philosophical Society, the American Society of Naval Architects and Marine Engineers, the American Academy of The Fine Arts, the University Club of Philadelphia, the City Club, Contemporary Club, Pennsylvania Society of Sons of the Revolution, and the New England Society of Pennsylvania. He was one of the founders of the Philadelphia Engineers' Club, had served on the Board of Managers and later as vice-president of The Franklin Institute, was a former president of the Chamber of Commerce of Philadelphia, and from 1908 until his death was one of the three State Commissioners of Navigation for the Delaware River.

JAMES F. SHACKLEFORD

James F. Shackleford, assistant superintendent gasoline plants, Standard Oil Co. of Indiana, Drumright, Ind., died on April 24, 1922. Mr. Shackelford was born in Columbus, Ga., on August 17, 1888. He was educated in the schools of Savannah, Ga. From 1913 to 1917 he was employed in the oil fields of Oklahoma, part of that time being spent with the Shaffer Oil & Gas Co. in the general construction and operation of gasoline plants. In 1917 he became connected with the Standard Oil Co. of Indiana and at the time of his death was assistant superintendent, in charge of five casinghead plants in the Drumright region.

Mr. Shackelford became an associate member of the Society in 1921.

WILLIAM DURWOOD SHERMAN

William D. Sherman, president and consulting engineer of the Industrial Engineering Co., Los Angeles, Cal., died on October 11, 1922. Mr. Sherman was born on August 15, 1859, in Dixon, Ill., where he received his early education. From 1871 to 1883 he served his apprenticeship in the manufacture of agricultural machinery in Dixon. He was then employed by the Union Hydraulic Drain Tile Co., Omaha, Neb., where he gained his drafting-room experience. He resigned to become associated with the Wilson & Baillie Manufacturing Co., Brooklyn, N. Y.

While with this firm he designed and perfected machinery for the manufacture of cement, egg-shaped and round sewer pipe and also designed a very complete conveyor system. He designed, erected and operated the system for a filter plant in Albany, N. Y., and as superintendent and mechanical engineer built a mixer and engines for the system of concrete walls for the greater part of the Wallabout Basin, Brooklyn. For the past ten years Mr. Sherman had been located in Los Angeles as president of the Industrial Engineering Co.

Mr. Sherman became a member of the Society in 1907.

WILLIAM E. SNYDER

William E. Snyder, mechanical engineer of the American Steel & Wire Co., Pittsburgh, Pa., died on June 24, 1922. Mr. Snyder was born on June 15, 1868, in Allegheny County, Pa. He was educated in the old Duquesne University, Curry University and Pennsylvania State College, from which he was graduated in 1896 with the degree of B.S. in mechanical engineering, receiving his M.E. degree in 1900.

Upon graduation he became associated with the Shoenberger Steel Co., Pittsburgh, where he gained his drawing-room and shop experience, then becoming assistant master mechanic of the same company. In 1899 Mr. Snyder accepted a position with the American Steel & Wire Co. as mechanical and electrical expert. A little later he was given the position of power engineer in the Pittsburgh district of the concern and in 1902 was appointed mechanical engineer, which position he held at the time of his death.

For five years Mr. Snyder was secretary of the Committee of Steam Engineers formed in the U. S. Steel Corporation by a number of the constituent companies who appointed representatives for this work. The purpose of the Committee was to make recommendations in the various plants of the corporation which might lead to improvements in economy of various kinds, but particularly that of fuel.

In the early part of 1908 Mr. Snyder was delegated by the American Steel & Wire Co. to investigate the utilization of blast-furnace and coke-oven gas in Europe, where he visited a large number of manufacturing plants in France, Germany, Austria, Belgium, Holland, Switzerland, Italy and also England and Wales.

Mr. Snyder became a member of the Society in 1900. He was a past

vice-president of the Engineers' Society of Western Pennsylvania and a member of the American Iron and Steel Institute. He also belonged to the Masonic Order.

WILLIAM NORTON STEVENS

William N. Stevens, an officer of the Stevens-Aylsworth Co., Inc., New York, N. Y., died on December 28, 1922. Mr. Stevens was born on January 11, 1863, in Brooklyn, N. Y., where he received his early education. He was graduated from the Stevens Institute of Technology in 1885.

He gained his shop experience at the hydraulic works of the Worthington Pump & Machinery Corporation in Brooklyn, N. Y., and later served as engineer with the J. G. White Co., Ford, Bacon & Davis, the New York Edison Co., and the Interborough Rapid Transit Co., all of New York City. He was particularly identified with the construction of the 76th Street power house and the earlier New York subways.

From 1910 until his death he was an officer in the Stevens-Aylsworth Co., Inc., and in his work for this company designed many improvements in chemical apparatus and was the holder of a number of valuable patents.

Mr. Stevens became a junior member of the Society in 1886. He was also a member of the American Geographic Society.

ARTHUR BOWERS STITZER

Arthur B. Stitzer, chief engineer of the Republic Railway & Light Co., consulting and contracting engineers, New York City, died on May 26, 1922. Mr. Stitzer was born in Hackettstown, N. J., in July, 1877. He was graduated from the University of Pennsylvania in 1899 with the degree of B.S. in electrical engineering. In 1909 he received his E.E. dgree.

From 1899 to 1910 he was associated with the Union Traction Co. which later became the Philadelphia Rapid Transit Co., Philadelphia, Pa. He entered their employ as a draftsman. In 1903 he was advanced to the position of assistant engineer and in 1905 was appointed electrical engineer of the company.

In 1910 he resigned from that concern to become connected with the firm of Ford, Bacon & Davis as electrical engineer in charge of design and construction in connection with the production, transmission and distribution of power. In 1917 he accepted the position of chief engineer with the Republic Railway & Light Co.

Mr. Stitzer became a member of the Society in 1917.

J. WHEATON STONE

J. Wheaton Stone, general manager and treasurer of the Eastern Machinery Co., New Haven, Conn., died on February 22, 1922. Mr. Stone was born on August 18, 1859, in Waterbury, Conn. He was educated in the schools of that city and later took a course in mechanical drawing.

He served his apprenticeship from 1877 to 1880 with J. E. Gaylord, where his experience covered planning and estimating as well as iron-foundry

and machine-shop work. In 1880 he was promoted to the position of foreman which position he held for two years when he took charge of the foundry department of the Collins Co., Collinsville, Conn., manufacturers of iron, brass and steel castings. In 1884 Mr. Stone became general foreman of the McLagon Foundry Co., New Haven, Conn., where he had entire charge of the designing, building and installing of power-transmission machinery, etc., resigning in 1895 to become general manager and treasurer of the Eastern Machinery Co.

Mr. Stone became a member of the Society in 1898. He was a member of the Masonic Order and belonged to several New Haven Clubs.

LOUIS E. STROTHMAN

Louis E. Strothman, Vice-President of the Society, died at his home in Milwaukee, Wisconsin, on May 8, 1922, after an illness of some months. Mr. Strothman was born in Milwaukee in 1879 and received his education in the public schools there and at St. John's Military Academy. From 1899 to 1902 he served as draftsman for several Milwaukee concerns, and then entered the employ of the Allis-Chalmers Manufacturing Co., with which he was associated in various capacities until August, 1919. At that time he became vice-president and general manager of the Richardson-Phenix Co., which position he held until his death.

Mr. Strothman had an extensive and varied engineering experience. He was a member of a number of engineering societies, including the American Society of Civil Engineers, the American Waterworks Association, the National Association of Stationary Engineers, and the Engineers Society of Milwaukee, of which latter he was president in 1916–17. In 1916 he was appointed a member of the board of directors of the Organization for National Preparedness for the state of Wisconsin, and was also appointed an associate member of the Naval Consulting Board.

He became a member of The American Society of Mechanical Engineers in 1909 and has taken an active interest in its affairs. In 1915–16 he was chairman of the Milwaukee Section. He served as chairman of the 1917 Nominating Committee and has since that time been a member of the Main Committee on Power Test Codes and chairman of the individual committee on Reciprocating Displacement Pumps. In 1919 he was appointed by President Cooley to represent the Society in company with himself on the National Industrial Conference Board. Later in the same year he was elected manager of the Society to fill a vacancy for a year. In 1921 he was elected Vice-President of the Society, which office he was holding at the time of his death.

The Council of the Society, at the Spring Meeting at Atlanta, Ga., received the news of Mr. Strothman's death with great regret, and voted to appoint a special committee to draw up resolutions to be entered upon the records of the Society and to be sent to his family.

From a close friend of Mr. Strothman's, Henry A. Allen, consulting engineer, Chicago, we have received the following appreciation: "Louis E. Strothman was a gentleman and an excellent engineer, always striving to

increase his knowledge and to better the engineering profession. When manager of the pumping-engine department of the Allis-Chalmers Co. about twenty years ago, I had the opportunity of advancing him from the drawing board to the desk, where he handled the estimates and correspondence dealing with centrifugal pumps. He was very efficient and his rise from that time on to manager of the department was deserved and steady. His death was a distinct loss to the engineering profession, and to me and many others means the loss of another long-time dear friend."

GILBERT CARL SUNDBERG

Gilbert C. Sundberg, assistant master mechanic with the Wyman-Gordon Co., Worcester, Mass., died on October 25, 1922. Mr. Sundberg was born on July 1, 1890, in Worcester, Mass., where he received his early education. Later he attended the Wentworth Institute of Boston from which he was graduated in 1914, in the machine construction and tool-design course.

He obtained his shop experience with the Wyman-Gordon Co., and his early experience in designing with the firm of Wells Brothers, Greenfield, Mass., and with the Winchester Repeating Arms Co., New Haven, Conn. In 1918 he returned to the Wyman-Gordon Co. as designing engineer, later being advanced to the position of assistant master mechanic.

Mr. Sundberg became a junior member of the Society in 1920. He belonged to the Masonic Order.

KNOX TAYLOR

Knox Taylor, president since 1910 of the Taylor-Wharton Iron & Steel Co., High Bridge, N. J., died on April 4, 1922. Mr. Taylor was born on October 19, 1873, in High Bridge and received his early education in the schools of Chester, N. J. Later he attended the Hill School at Pottstown, Pa., and the Model School at Trenton, N. J. He was graduated from Princeton University in 1895 with the degree of B.S. and his first employment was with Waddingham & Taylor, surveying and prospecting in New Mexico. A year later he became connected with the Philadelphia & Idaho Mining Co., Ketchum, Idaho, where he was located until 1902 as manager of the properties.

In 1902 he returned to the East to enter the employ of the Taylor Iron & Steel Co., where he started at the bottom and worked up through the various departments, becoming general manager in 1905 and president of the company in 1910. From the combination of William Wharton, Jr., & Co., Inc., Philadelphia Roll & Machine Co., Tioga Iron & Steel Co. with the Taylor Iron & Steel Co. grew the present organization. Each subsidiary retained its identity and Mr. Taylor was president of each.

Deeply interested in civic affairs of High Bridge, he was responsible for the formation of the Boy Scouts, the Y. M. C. A., and was interested in the bank and the home newspaper. During the War, in addition to the work of his company's own contracts, he helped supply railway-track ma-

terial for the A.E.F., representing in the dealings with the Government all of the principal manufacturers of this class of product.

Mr. Taylor was a member of a large number of organizations and clubs, among which were the American Institute of Mining and Metallurgical Engineers, the American Chemical Society, Society for the Promotion of Engineering Education, American Society for Testing Materials, Iron and Steel Institute, England; he was vice-president of the Railway Business Association, a life trustee of Princeton University, a member of the Engineers' Club of New York, and a director of the Delaware & Bound Brook Railway. He became a member of the Society in 1918.

WINTHROP GORDON THOMAS

Winthrop G. Thomas, vice-president of the Batavia Service Corporation, New York City, died on January 31, 1922. Mr. Thomas was born on September 14, 1892, in New York City. He studied for three years in the Massachusetts Institute of Technology. He was formerly associated with the Baltimore & Ohio Railroad as assistant efficiency engineer and with the Western Electric Co. as special sales representative and later as electrical engineer.

Mr. Thomas became a junior member of the Society in 1916.

THEODORE THURN

Theodore Thurn, engineer in charge of steam-turbine construction for the Japan office of the General Electric Co., died on November 9, 1922, in Geneva, Switzerland. Mr. Thurn was born in Fort Lee, N. J., in September, 1871. He was educated in the Royal Technologic Institute, Dresden, Germany, receiving his M.E. degree in 1898.

He gained his shop experience with Gebr. Koerting, Hanover, Germany, and the Dresden Gasmotor Fabrik, Dresden. Upon coming to this country he entered the drafting room of the General Electric Co. in Schenectady, N. Y., and later worked through the steam-turbine shops. For six years he had charge of turbine-construction work and general engineering for the General Electric Co. both in this country and Japan. For the last fifteen years he was associated mainly with the Yokohama, Japan, office of the company, handling the steam-turbine end of the business, for the promotion of which in Japan he was largely responsible, having superintended and had in his charge Curtis steam turbines installed by the G. E. Co. amounting in capacity to over 300,000 kw.

Mr. Thurn became a junior member of the Society in 1904 and was advanced to full membership in 1909. He belonged also to the Japan Society for Mechanical Engineers, in Tokyo, the American-Japan Society, Tokyo, the Tokyo Club, the American Association at Tokyo and the Yokohama United Club.

C. LOTHROP TOWER

C. Lothrop Tower, sales engineer in the Boston office of the Babcock & Wilcox Co., died on July 19, 1922. Mr. Tower was born on June 20, 1887,

in Auburndale, Mass. He attended the Newton, Mass., high school, and later Cornell University where, after three years' study toward his M.E. degree, he was taken ill and forced to leave.

He did not return to the University but entered the employ of Lee, Higginson & Co., Boston, Mass., where he engaged in statistical work in their bond department. At the end of three years he resigned from this position to become a partner in the firm of A. F. Carva & Co., manfacturers of portable electric tools, with charge of the design and production end of the business. In 1916 he became connected with the Stone & Webster Co., Boston, as draftsman and assistant engineer, resigning to become associated with the Babcock & Wilcox Co., as sales engineer in their Boston office.

Mr. Tower became a member of the Society in 1921.

CHARLES WELLINGTON WALKER

Charles W. Walker, works manager of the Walker Silent Typewriter Co., Bridgeport, Conn., died in June, 1922. Mr. Walker was born on August 6, 1856, in Worcester, Mass. He was educated in the State Normal School, Orangeville, Pa.

From 1893 to 1906 he was engaged as expert and inventor by the Union Typewriter Co., Bridgeport. From 1917 to 1920 he served as engineer with the Remington Arms Co., also in Bridgeport. The following year he became works manager of the Walker Silent Typewriter Co.

Mr. Walker became an associate member of the Society in 1921.

DRUID A. WALTON

Druid A. Walton, president C. J. Walton & Son, Louisville, Ky., died on October 30, 1922. Mr. Walton was born on September 24, 1864, in Louisville, Ky., where he received his early education. Later he attended Stevens Institute of Technology and was graduated in 1887 with the degree of M.E.

Immediately upon graduation Mr. Walton returned to Louisville and entered his father's boiler works with which he was associated to the time of his death. He worked his way up through the shops, holding various positions, to the appointment of general manager. Twelve years ago he became president of the firm, which position he held at the time of his death.

Mr. Walton became a member of the Society in 1920.

ERNEST D. WILLIAMS

Ernest D. Williams, turbine engineer in the Erie Works of the General Electric Co., Erie, Pa., died on March 25, 1922. Mr. Williams was born on September 7, 1883, in Martin's Ferry, Ohio. He was educated in the public schools and Ohio University from which he was graduated in 1905 with the degree of M.E.

During his summer vacations he worked in the shops of the Carnegie Steel Co. and also with the Puzzle Mining and Milling Co., Breckenridge, Col. Upon his graduation he became connected with the General Electric Co., Schenectady, N. Y., in the department for the experimental and com-

mercial testing of steam turbines. He was with this concern until the time of his death although not in the same office, having been transferred to the Erie Works of the company in 1915.

Mr. Williams became a junior member of the Society in 1905.

JAMES E. WILLITS

James E. Willits of the Thermal Efficiency Co., Kansas City, Kansas, died suddenly on June 20, 1922, from an accident in diving. Mr. Willits was born in Quincy, Ill., on June 23, 1891. He was educated in the public schools of Brookline, Mass., and attended Amherst College for one year.

In 1911 he became connected with the Worthington Pump & Machinery Corporation, working on installation and tests of condensers and auxiliaries at the Boston Edison and Boston elevated power plants. The following year he became resident superintendent for the same concern in charge of the erection and test of a waterworks pumping engine, installed at Garden City, N. Y. From 1912 to 1917 he held the position of sales engineer in the Kansas City office of the company, resigning at that time to enter the Service.

He was commissioned a first lieutenant in the Field Artillery of the Army and later was advanced to the rank of captain. He saw active service overseas, taking part in five major engagements and being gassed in one. He was honorably discharged from the Service in June, 1919, and returned to his position as sales engineer in the Kansas City office of the Worthington company.

Early in 1920 Mr. Willits accepted the position of district manager and engineer in charge of sales for the Merkle Machinery Co., Kansas City. He resigned from this company to open his own office as engineer and contractor in Tulsa, Okla. He had become associated with the Thermal Efficiency Co. just a short while before his death.

Mr. Willits became an associate member of the Society in 1921. He belonged to several clubs in Kansas City and was a member of the American Legion.

BENJAMIN F. BROWN

Benjamin F. Brown, president and manager of the Brown Bag Filling Machine Co., Fitchburg, Mass., died on February 21, 1922. Mr. Brown was born on December 8, 1849, in Lowell, Vt., where he received his early education. Later he attended Amherst College from which he was graduated in 1874 with the degree of A.B.

From the time of his graduation to 1890, Mr. Brown was engaged in the teaching profession in the following schools: Fitchburg high school from 1874 to 1880; the Day Street grammar school, Fitchburg, from 1880 to 1881; the Montpelier, Vt., high school from 1883 to 1885; and the Gibson School, Boston, from 1885 to 1890.

In 1890 Mr. Brown started the Brown Bag Filling Machine Co. which was finally organized two years later with him as president and general manager. This company engaged in the development and perfection of machines for handling flat paper bags and filling them with measured quantities of any designated material.

Mr. Brown became a member of the Society in 1916.

RESEARCH REPORT ON FLUID METERS

As the result of many years of study, the Special Research Committee on Fluid Meters, R. J. S. Pigott, Chairman, presented its first report to the Society in December 1922. This report takes the form of a reference book on flow meters of all kinds. It contains not only such practical instruction and information, including formulas, constants, etc., as may be needed by the actual or prospective user, but also more general information — the physical principles of design and operation — which may be useful to students, designing engineers, and inventors.

Part I treats of the general types of flow meters as well as the principles and methods involved and gives information which may, in many cases, be applicable to various commercial meters. In this part individual makes of instruments are not discussed in detail, but are referred to only incidentally or for illustrative purposes. The general physical principles are in the body of the text, while the derivation of formulas and the refinements of the theory involved have been placed in the appendices which accompany the reports.

In Part II will be found more detailed information concerning the practical use of all the flow meters now in common use. This information has been obtained from both makers and users and includes descriptions of commercial meters, and particulars with respect to operating characteristics, influence of installation, and the testing of meters. Since this report is too voluminous to be reprinted in Transactions, it has been printed as a separate pamphlet.

POWER TEST CODES

During the year 1922 the following Power Test Codes passed completely through the procedure which has been laid down for the formulation of these standard test codes. They will, therefore, be made available in pamphlet form during 1923.

Definitions and Values, R. J. S. Pigott, Chairman of Committee.

Displacement Compressors and Blowers, Paul Diserens, Chairman of Committee.

Feedwater Heaters, Geo. A. Orrok, Chairman of Committee.

Hydraulic Power Plants, W. F. Uhl, Chairman of Committee.

These four test codes, together with the two which were referred to in Vol. 43 (1921), consititute the work which has been actually completed by the Committee of 125 specialists which, in 1919, began the revision of the A.S.M.E. Power Test Codes of 1915. When completed these revised codes will number 18 and will be supplemented by the three codes on General Instructions, Definitions and Values, and Instruments and Apparatus.

No. 1883

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[1] See MECHANICAL ENGINEERING, April 1923
[2] See MECHANICAL ENGINEERING, April 1923
[3] To be published in TRANSACTIONS, vol. 45.

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